DESIGN AND MATHEMATICAL MODELLING OF AN EVAPORATIVE COOLING SYSTEM — A CASE STUDY

by

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INDIAN INSTITUTE OF TECHNOLOGY KANPUR

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AND
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OF AN
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— A CASE STUDY

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CERTIFICATE

It is certified that the work contained in the thesis entitled **DESIGN AND MATHEMATICAL MODELLING OF AN EVAPORATIVE COOLING SYSTEM —A CASE STUDY**, by Capt.C STEPHENS, has been carried out under my supervision and that this work has not been submitted elsewhere for a degree.

29/3/98

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March,1998

Dedicated to -

My Beloved Wife

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Synopsis

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Design And Mathematical Modelling Of An Evaporative Cooling System — A Case Study

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Evaporatively cooled air ventilation for airconditioning (ECAVA) is an energy efficient environmental friendly method as it uses only water as the working fluid. Though ECAVA is not a substitute of mechanical air conditioning,yet it renders quite a reasonable level of comfort for relieving the thermal strain of human beings and that too without using ozone depleting chemicals.

A mathematical model developed by Verma was modified for calculations of the heat gain. by a building of a given geometry. Thereafter the ventilation air is calculated for a preassigned temperature rise of evaporatively cooled air. The salient feature of the present mathematical model is that it has provisions for making various changes in parameters through sets of input commands. The model was applied for a three storey production shop of a company at Kanpur involved in

manufacture of scooters and other components. The volume flow rates of air were calculated for all the three floors. For the known volume of air for the building the sizes of axial-flow fans were chosen from the existing catalogues of standard manufacturers. To reduce energy requirements and to have less vibration and noise the low speed axial-flow fan was identified. The electrtic energy consumption was further reduced by introducing humidity controller. The same not only saves electric energy, but also reduces wear and tear of pump, heating of motor and noise drastically. Solenoid valves were also incorporated to help conserve electric energy.

Since there are no laid down norms in India specifying the details of density for wood fibres used in evaporative coolers,the ASHRE guideline was followed. To keep the cooler pads always wet water distribution was ascertained by conducting experiments for the water flow rates to help the designer obtain practical ranges of discharge coefficients.

The air distribution ducts were designed on the basis of equal friction pressure drop method. Care has been taken to balance the pressure in main branches as far as possible. To calculate various duct dimensions,a computer programme was developed. Wherever structural constructions came,the duct dimensions were changed maintaining the same diameter of the ducts. This approach was adopted to ensure equal friction drop for the system.

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Nomenclature

A area of walls/roof, [m^2]

A_{door} area of door, [m^2]

A_{sun} glass area directly exposed to sun, [m^2]

A_{win} total glass area of window, [m^2]

c specific heat capacity of walls/roof, [kJ/kg-K]

CLF cooling load factor

d declination angle

F_{sg} angle factor between surface and ground

F_{ss} angle factor between surface and sky

h hour angle

h_o outside film coefficient, [$W/m^2.K$]

h_i inside film coefficient, [$W/m^2.K$]

H height of window, [m]

I_d diffused solar radiation, [W/m^2]

I_{dg} diffused solar radiation reflected from ground, [W/m^2]

I_{ds} diffused solar radiation reflected from clear sky, [W/m^2]

I_D direct solar radiation, [W/m^2]

I_{DN} direct normal radiation, [W/m^2]

I_t total solar radiation, [W/m^2]

I_{tH} total solar radiation falling on ground, [W/m^2]

I_r reflected solar radiation, [W/m^2]

k thermal conductivity of walls/roof, [W/m-K]

k_g thermal conductivity of window glass, [W/m-K]

l latitude of the place

Lifiltration/ventilation air in litre per second

no number of people in the conditioned space

N_{ach} number of air changes per hour

$\dot{Q}_{al}$ latent load due to appliances, [W]

$\dot{Q}_{as}$ sensible load due to appliances, [W]

$\dot{Q}_{door}$ heat transfer through door, [W]

$\dot{Q}_{glass}$ heat transfer through glass, [W]

$\dot{Q}_{il}$ latent load due to infiltration, [W]

$\dot{Q}_{is}$ sensible load due to infiltration, [W]

$\dot{Q}_{latent}$ total latent load, [W]

$\dot{Q}_{ls}$ sensible load due to lighting, [W]

$\dot{Q}_{ol}$ latent load due to occupancy, [W]

$\dot{Q}_{os}$ sensible load due to occupancy, [W]

$\dot{Q}_{ps}$ sensible load due to power equipments, [W]

$\dot{Q}_{sensible}$ total sensible load, [W]

$\dot{Q}_{ss}$ sensible solar load, [W]

$\dot{Q}_{structural}$ heat transfer through walls/roof, [W]

$\dot{Q}_{total}$ total cooling load, [W]

$\dot{Q}_{vl}$ latent load due to ventilation, [W]

$\dot{Q}_{vs}$ sensible load due to ventilation, [W]

$(\dot{Q}_{roof})_{with\ evap.}$heat transfer through roof with surface evaporation, [W]

$(\dot{Q}_{total})_{with\ evap.}$total cooling load with surface evaporation, [W]

RH_i inside relative humidity, [%]

S_H shadow height due to horizontal projection, [m]

S_w shadow width due to vertical projection, [m]

t_g temperature of glass sheet, [^oC]

t_{gi} temperature of inner surface of glass sheet, [^oC]

t_{go} temperature of outer surface of glass sheet, [^oC]

t_i inside air temperature, [^oC]

t_o outside air temperature, [^oC]

t_{si} inside surface temperature, [^oC]

t_{so} outside surface temperature, [^oC]

T_{db} dry-bulb temperature of outside air, [^oC]

T_{max} maximum air temperature of a day, [^oC]

T_{min} minimum air temperature of a day, [^oC]

T_{wb} wet-bulb temperature of outside air, [^oC]

T_{sol} sol-air temperature, [^oC]

U overall film coefficient, [$W/m^2.K$]

V_i inside air velocity, [m/s]

V_o outside air velocity, [m/s]

V_{room} volume of room, [m^3]

W width of window, [m]

$WB_{depression}$ wet-bulb depression, [^oC]

β solar altitude angle, [radian]

θ solar incidence angle, [radian]

ϕ solar azimuth angle, [radian]

ψ surface azimuth angle, [radian]

γ surface-solar azimuth angle, [radian]

Σ surface tilt angle from horizontal, [radian]

Ω profile angle, [radian]

ϵ emissivity of the surface

η_{se} saturation efficiency

α thermal diffusivity of walls/roof

α_d absorptivity of glass for diffused radiation

α_D absorptivity of glass for direct radiation

α_s absorptivity of surface

τ_d transmissivity of glass for diffused radiation

τ_D transmissivity of glass for direct radiation

ρ density of walls/roof, [kg/m^3]

ϱ_g reflectance of foreground

ΔT difference between inside and outside temperatures, [^oC]

Δx thickness of walls/roof, [m]

$\Delta\tau$ time interval, [s]

$\Delta\omega$ difference of inside and outside specific humidities

Chapter 1

Introduction

1.1 Description

According to the Oxford Dictionary the art of ventilating is,"to expose to fresh air", and to,"cause air to circulate freely in an enclosed space". The term general ventilation has acquired a special meaning in contradiction to 'local exhaust ventilation', and refers to the commonly understood process in which the space is constantly flushed with fresh or humidified out door air.

The object of air distribution in a ventilation and air conditioning system is to create the proper combination of temperature, humidity and air motion in the conditioned room for obtaining comfort conditions. The standard limit for an effective draft temperature', has been established, for this purpose. The same consists of air temperature, air motion, relative humidity and there physiological effects on the human body[1]. Any variations in the accepted standards of one of these elements or lack of uniform conditions in the same part of the space as room air temperature variations (horozontally, vertically or both) and excessive air motion (draft) will lead to discomfort.

In India, Malhotra [2] has obtained the effective temperatures for hot & humid as well as hot & dry climates based on recording of 75 % votes by subjects about their sensation in different working environments. Table 1.1 illustrates

the comfort range found by him

Table 1.1: Recommended Effective Temperatures [°C].

Sl. No.	Level of Comfort	Effective Temperature	
		Hot humid climate	Hot dry climate
1.	Warm and unpleasant	27.0	26.0 - 28.3
2.	Comfortable and pleasant (upper level)	24.0 - 25.0	24.4 - 26.6
3.	Comfortable and pleasant (lower level)	22.0 - 22.5	21.1 - 24.3

Table 1.1 shows that high effective temperature of 27 °C for subjects of hot-humid climate as compared to about 22 °C of existing practice. The energy requirement is reduced significantly due to the higher temperature. The thermal comfort requirements for hot-humid and hot-dry climates have been studied elaborately by Kimura and Tanabe[3] with a view to energy conservation in buildings. The results are presented in Table 1.2 having various combinations of human comfort parameters. The higher is the inside condition, the lower is the thermal potential across the building structure. Hence energy requirement gets reduced considerably.

Table 1.2: Parameters for Human Comfort [3].

Sl. No.	Experimental Condition	Velocity(m/s)
1.	27 °C, $\phi = 50$ %	0.5
2.	29 °C, $\phi = 50$ %	1.2
3.	31 °C, $\phi = 50$ %	1.6

If high comfort condition is permitted, even evaporative cooling becomes feasible. In fact it was this very concept due to which the concept of hybrid air conditioning was envisaged. The hybrid system is a process based air conditioning system as the comfort condition in a hot-dry climate is achieved by

evaporative cooling rather than the wrongly used mechanical airconditioner. The mechanical airconditioner is not only the wrong system for maintenance of comfort condition in hot-dry climate, but also takes energy about 7 times that of an evaporative cooling system[4].

It is a well known fact that the evaporative airconditioning is not a substitute of mechanical airconditioning, but it renders reasonable degree of comfort and helps relieve thermal strain to a great extent. That is why these days, ventilation with evaporative cooling is becoming popular to increase productivity in industrial organisations.

For maintenance of desirable temperature, the circulation should be high enough to have less temperature rise. For this purpose total heat gain of the confined space or building is calculated. Thereafter the ventilation air is estimated on the basis of energy balance between the heat gain of the building and cooling to be produced by evaporatively cooled air.

The cooling heat gain calculations are based on the design outside maximum and minimum temperatures. However several researchers have represented outside daily hourly temperature by sinusoidal function. On the other hand the ASHRAE suggests an hourly temperature as the percentage of daily range for cooling heat gain calculations [5]. It gives the hourly multiplying factor in a tabular form for calculation by the designer. The ASHRAE approach is similar to using sinusoidal temperature variations based on maximum and minimum values. Because in both methods only maximum and minimum daily temperatures are needed and they dispense with the complete temperature-time history of the place for which heat gain is to be calculated. The hourly temperatures have been calculated using the method of ASHRAE and sinusoidal temperature variations approach. In either method the predicted values differ by only a small magnitude about 2 - 3 %. The predictions for a few hours by the latter is compensated by the over prediction by the same for the rest of the period. The same is not going to make much changes in the final predictions.

1.2 Energy Conservation And Global Warming

The era of industrial revolution has arroused a need to be inquisitive as to why so many governments in the world have focused together to address the problem of global warming? The answers are not absolutely clear, but it is apparent that mankind's activities are perturbing the natural variations of carbon, nitrogen and chlorine compounds in the atmosphere and many governments especially in the developed world, have decided to take action in a big way.

In late 1973 the Organisation of Petroleum Exporting Countries (OPEC) extended the life of their income source by restricting oil production and allowing demand to force the price in excess of $ 2.50 per barrel for the major importing countries. This action led rapidly to revaluation of all forms of energy. The developed world focussed first on turning of lights and equipment when not required, drastically cutting back ventilation air quantities, and other similar conservation measures.

It was not until a second, a much more substantial increase in price of oil in 1978, that the developed nations resorted to design and development of efficient goods and services. Evaporative airconditioning technologies even though they are still underutilised and often even unknown in many parts of the world, provide comfort cooling throughout many arid zones of the world, as well as relief cooling for commercial and industrial applications such as green houses, laundries, warehouses, factories and poultry houses[9].

1.3 Three Important "E's".

. They stand for :

- *Energy*(First E) :Energy demand rising to sustain various industrial activities and other applications.
- *Environment*(Second E) :Increasing energy use renders environment degradiation below allowed limit due to release of global warming and polluting gases, etc.
- *Economy*(Third E) :Investment renders recovery to sustain existing need and future requirement.

A few decades back the main aim was to meet the energy needs as far as possible. Hence, the first E dominated. However, the increasing energy use caused tremendous increase in release of particulate and polluting gases in addition to global warming gases. In India alone the carbon dioxide release from thermal power plants turns out to be over 800 million tonnes per year and in the world about 7 billion tonnes per year. Four main gases: nitrogen, oxygen, argon and carbon dioxide form about 99.99% of the atmosphere, with traces of rare gases like neon, methane, krypton, helium, xenon, hydrogen, carbon monoxide, radon and ozone. Of these gases CO_2 , water vapour and methane are the green house gases (GHG). These gases allow the incoming shortwave radiation from the sun to the earth's surface while the GHG's absorb partly the reradiated low energy radiation, thereby the environmental temperature near the earths surface remains at an average temperature of 288K as against 259K which it would have remained. Since the GHG's surpassed the accepted limits of environmental quality and the degraded environmental quality started affecting health of living beings including plants and crops. Hence, energy demand for sustaining various activities cannot be seen in isolation of pollution, globalwarming and ozone depletion. The ash released from the thermal power plants also cause pollution as well [6].

The pollution level has gone so high that the environmentalists are taking action against the reckless growth of thermal power plants. The seriousness of the environmental problems has also been accepted by the world bank. The same is reflected from the fact that the World Bank keeps the environmental

aspect mandatory for release of funds in support of development of new power plants. Thus, the second E stands for environmental pollution.

Refrigeration systems cause two types of contribution to the global warming effect :

- Direct Global Warming Potential (DGWP).
- Indirect Global Warming Potential (IGWP).

1. DGWP: Taking the reference base as the GWP of CO_2 over a 100 years period:
   (a) The GWP of R134a is 1200 times to that of CO_2.
   (b) The GWP of R404A is 3500 times to that of CO_2.
2. Total Equivalent Global Warming Impact (TEWI): The Total Equivalent Global Warming Impact includes both GWP, under conditions of refrigerant loss and recovery, expressed as ($GWP\times$ total emission), and the "additional" (so called indirect) global warming from CO_2 produced throughout the system's service life by generating the electricity, or the direct combustion, needed to run the equipment. $TEWI$ is useful for showing whether the direct effect or the indirect effect is dominant in a particular system [9] .

   $TEWI$ is calculated from the equation:

   $$TEWI = (GWP) \times (M) + (a) \times (B) \tag{1.1}$$

   where,

   $GWP = GWP$ of refrigerant relative to CO_2 and $GWP(CO_2) = 1$

   M = Total mass of refrigerant released (kg).

   a = Amount of CO_2 released in generating electricity (kg CO_2 per KW)

   B = energy consumption of the system in its life time.

$TEWI$ is much more reliable indicator of global warming for a particular system than is GWP.

CFC refrigerants being responsible for ozone depletion/ozone hole. The Evaporative Cooling Airconditioning (ECA) system requires only water as the working medium. Hence, the system does not release global warming gas like CO_2 nor polluting gases. Even this system takes only one-fifth to one-eighth of the electric power as compared to the vapour compression system . Hence, the systems TEWI is very less for ECA. When water-vapour compression system is used,in this case also green house gas emissions are absent. But, water-vapour itself is a global warming gas.

With the increasing competetiveness in business, to compete with the other quicker methods of cooling, the investment in the evaporative cooling system renders much recovery to sustain future requirement. Hence, evaporatively cooled air for airconditioning (ECAA) has acquired great signifiance as the investment for power for ECAA is much smaller than that for the vapour-compression system rendering saving in revenue for further upkeep, upgradation and instillation of new power plants to meet increasing demands in future. This aspect is very important. Evaporative cooling systems are more inexpensive to purchase and easier to operate as compared to the vapour-compression airconditioners.

1.4 Present Work

A computer programme is modified for calculation of cooling loads for a manufacturing shop in the Industrial Estate, Kanpur. The unsteady nature of the outside temperature variation and the incident solar radiation is included by considering a decrement factor and time lag factor. By this approach the calculations are simplified significantly. The details are available in Sec.2.3. Based on the heat gain for the building the volume of ventilation air required per floor of the three storeyed manufacturing shop was worked out, keping the space limitations and user's needs in

mind. Depending on the volume of air flow per floor a computer programme for duct sizing was developed keeping the distribution of air to all parts of the building in view.

Finally, evaporative cooling systems were designed and their layout was suggested for instillation on top of the building due to space restrictions. Experiments were conducted for evaluation of discharge coefficient for the holes in the trough used for water distribution to the evaporative coolers. To keep energy conserved and without affecting evaporative cooler performance electronic timers have been recommended for all three evaporative cooling systems. The air handling capacities for evaporative cooling systems for ground, first and top floors turned out to be, 80,000m^3/h, 40,000m^3/h and 14,400m^3/h, respectively.

Chapter 2

Cooling Load Calculation

In this chapter,the expressions used for evaluation of the various individual heat gains are first presented followed by a summation of all loads to give the total heat gained by the structure and finally the expression for the volume of ventilation air are presented.

2.1 Different Heat Sources :

The cooling load for the evaporative air conditioning systems for summer, in general, comprises:

- *Sensible Heat Load*
- *Latent Heat Load*

The following constitute the sensible heat load [10] :

(a) Solar Radiation : It is again divided into two categories:

i. Direct Solar Radiation : It is the heat load which results from the transmission of solar radiation through glass windows and ventilators.

ii. Indirect Solar Radiation : A part of the solar radiation falling over the walls and roofs gets absorbed and then transmitted to the room by conduction and convection.

(b) Heat Transfer by Temperature Difference: The heat transfer through the exterior walls, roofs, floors, doors etc., takes place due to temperature difference between the surroundings and interior space.

(c) Heat Released from Occupants

(d) The electrical energy dissipation from various equipments and appliances add to the heat energy.

(e) Miscellaneous heat sources.

The sources which contribute to the latent-heat load are listed below:

(a) The latent heat load from the occupants;

(b) Moisture passing directly into the space through permeable walls where water-vapour pressure is high;

(c) The latent-heat load from cooking foods and from stored materials.

The cooling load arises from the following sources:

(a) External Sources :

  i. Heat transmission through barriers such as walls, doors, ceiling, floor, etc. being caused by the temperature difference existing on the two sides of the barrier.

  ii. The solar heat , absorbed by walls , roofs and windows exposed to radiation from the sun and transferred to the inside space.

  iii. Heat and moisture introduced to the conditioned space through infiltration and ventilation air.

(b) Internal Sources :

  i. Occupancy load, both sensible and latent

  ii. Lighting load

  iii. Power equipment load, e.g., fans, water pumps, etc.

  iv. Appliances load, both sensible and latent

2.2 Need for Hourly Cooling Load Calculations

It is a well known fact that the ambient air temperature is not constant during day, and accordingly the cooling load varies over a period of 24 hours. This is mainly caused by the variation in the solar intensity falling on the earth surface. The cooling heat gain calculation is further complicated by the fact that a wall has thermal capacity, due to which a certain amount of heat passing through it is stored and is transmitted to the inside at some time latter. Therefore, calculation based on instantaneous heat transmission through structure without considering the thermal capacity of the wall, does not reflect the actual heat gained by the confined space. Moreover, the traditional methods of evaluating the cooling load involves various assumptions like the load from each component is constant in a day and maximum value of the cooling load is equal to the individual maximum, which, in general, is not correct, as the occurrence of maximum heat gain at particular time is not the occurrences of the other components. It is proper to adopt an hourly cooling load calculation to facilitate an economic selection of an evaporative air conditioning unit [5].

2.3 Structural Cooling Load :

2.3.1 Building Survey

The building survey incorporates the flowing particulars:

(a) Location of the building ,i.e., longitude and latitude

(b) Orientation

(c) Dimension of the building structure, such as length, breadth , height and thickness of each layer of the building material

(d) Composition of building material and their physical properties.

2.3.2 Hourly Outside Temperature Variation

Figures 2.1 to 2.4 show the hourly variation in outside temperature for Kanpur. The meteorological data reveals that the minimum temperature occurs just one or two hours before the sunshine while the maximum temperature occurs three to four hours after the solar noon [15]. Since, the temperature time history is available for only a few places, an hourly variation in temperature is predicted based on maximum and minimum temperatures.

The ASHRAE Handbook of Fundamentals [5] has outlined a procedure to predict the hourly temperature. In the present work, the same is followed.

$$t_o = T_{max} - DailyRange \times PercentageFactor/100 \qquad (2.1)$$

where, t_o=temperature at any time in ^oC

$DailyRange = T_{max} - T_{min}$

The percentage factor is taken for that hour for which outside temperature is to be calculated. These factors are tabulated in [5] and are reproduced here in Appendix 'A'.

The comparison of the temperature variation by theoretical with actual variation is shown in figures 2.1 to 2.4. The data have been taken from the average temperature of three years, i.e., 1982, 1983 and 1985 [15], and are reproduced here in Appendix 'B'.

2.3.3 Solar Heat Gain

Solar radiation forms a significant part of the cooling load for a building. The total radiation (I_t), reaching on terrestrial surface is the sum of the direct solar radiation (I_D), the diffused sky radiation (I_d) and the solar radiation reflected from the surrounding surface (I_r). The intensity of the direct component is the product of the direct normal radiation (I_{DN}) and the cosine of the incidence angle (θ) between the incoming solar rays

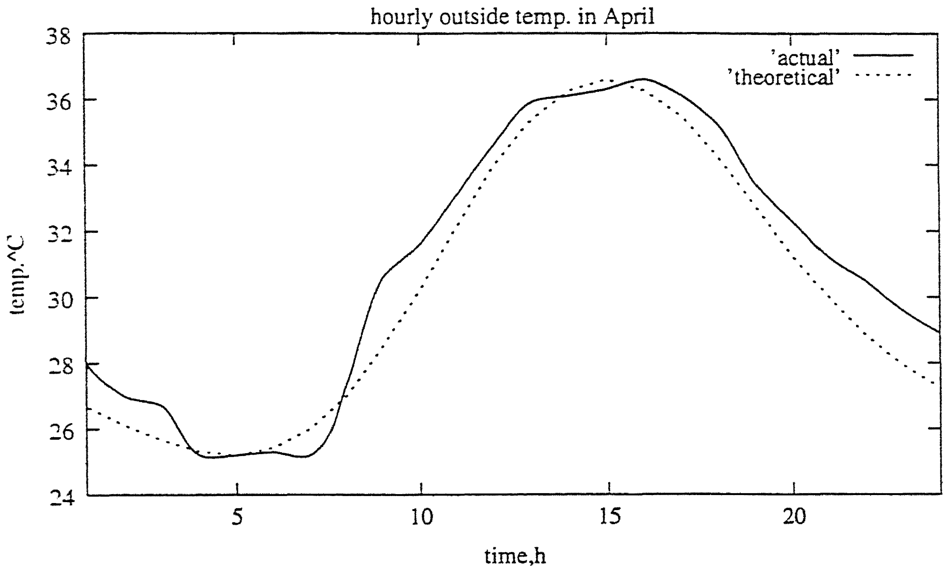

Figure 2.1: Hourly variation of actual and theoretical temperature in April.

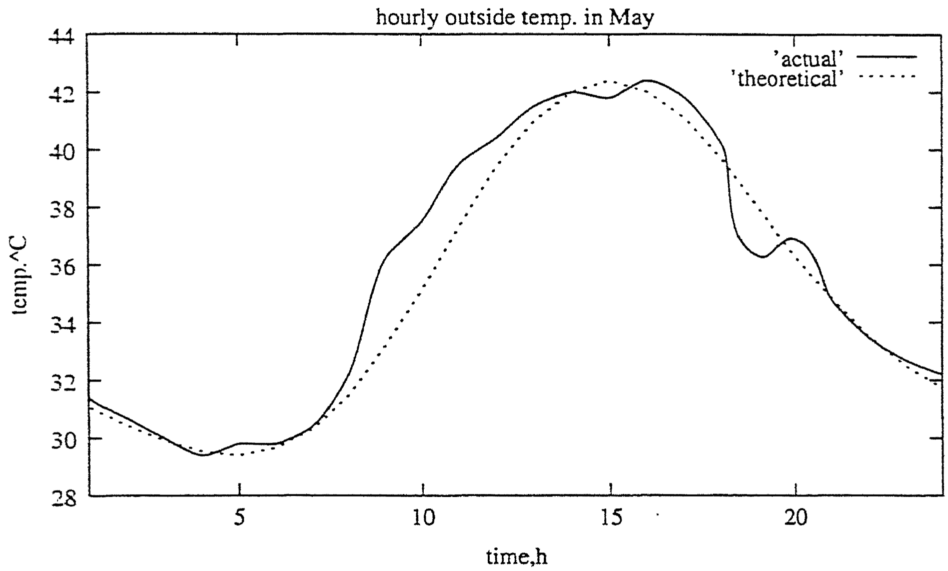

Figure 2.2: Hourly variation of actual and theoretical temperature in May.

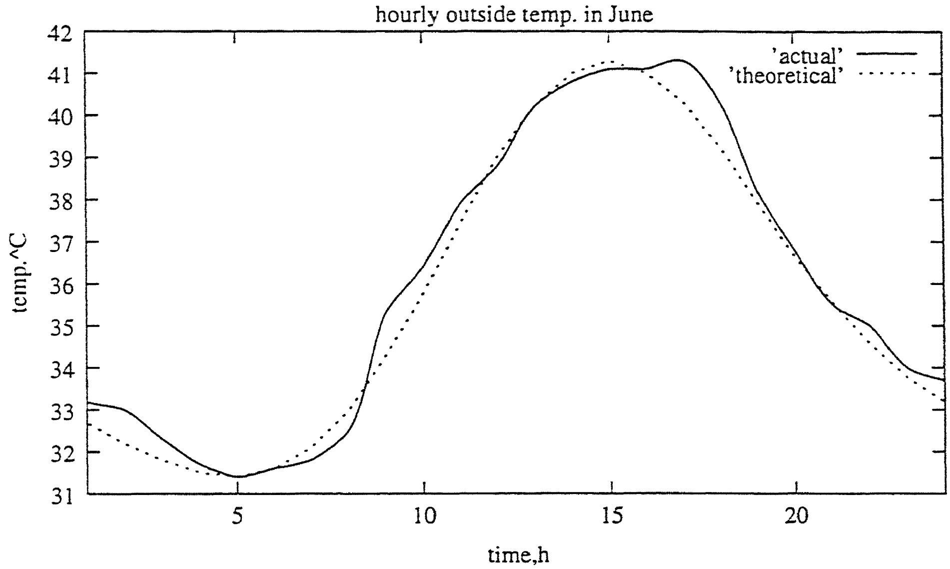

Figure 2.3: Hourly variation of actual and theoretical temperature in June.

and a line normal to the surface as shown in figure 2.5. Thus,

$$I_t = I_D + I_d + I_r \tag{2.2}$$

In the present analysis the reflected radiation (I_r) is neglected.

I_D is given as:

$$I_D = I_{DN} \times cos\theta \tag{2.3}$$

where, I_{DN} is expressed as :

$$I_{DN} = \frac{A}{exp(B/sin\beta)} \tag{2.4}$$

I_d consists two parts:

(a) diffused solar radiation from clear sky

$$I_{ds} = C.I_{DN}.F_{ss} \tag{2.5}$$

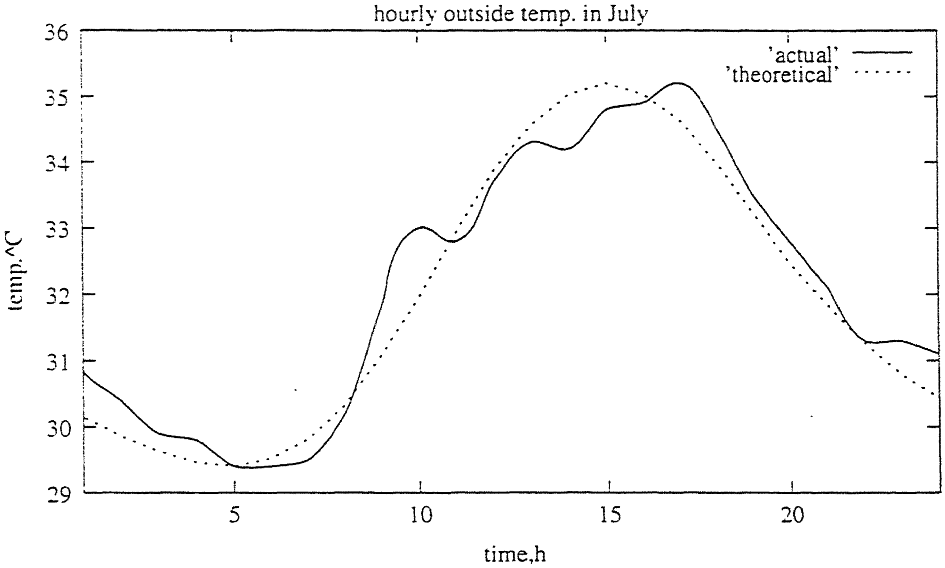

Figure 2.4: Hourly variation of actual and theoretical temperature in July.

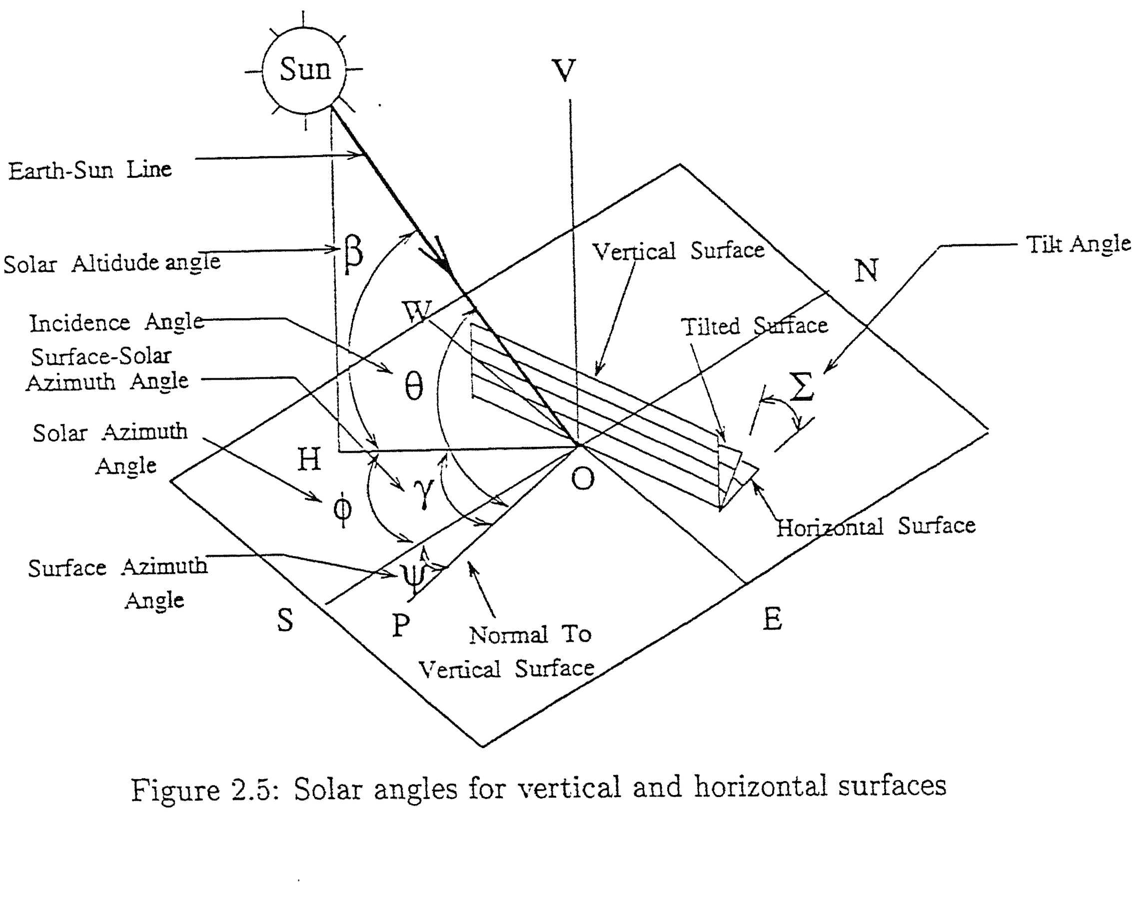

Figure 2.5: Solar angles for vertical and horizontal surfaces

where, F_{ss} is the angle factor between the surface and the sky, i.e., the fraction of short wave radiation emitted by the sky that reaches the tilted surface (dimensionless). F_{ss} is 0.5 for vertical surfaces and 1.0 for horizontal surfaces. For other surfaces,

$$F_{ss} = (1 + \cos\Sigma)/2 \tag{2.6}$$

where, Σ is the tilt angle from the horizontal plane(Fig 2.5).

(b) ground reflected diffused solar radiation

$$I_{dg} = I_{tH}.\varrho_g.F_{sg} \tag{2.7}$$

where, ϱ_g is the reflectance of foreground and F_{sg} is the angle factor between the surface and the ground (dimensionless), and $F_{sg} = 1 - F_{ss}$. I_{tH} represents the total solar radiation falling on the ground, and is given as$I_{tH} = I_{DN}(C + \sin\beta)$.

The value of A, B and C are taken from [5] and the same are reproduced in Appendix 'C' for the twenty first day of the each month.

2.3.4 Determination of Angles of Incidence

The sun's position in sky is most conveniently expressed in terms of solar altitude β, above the horizontal and solar azimuth angle ϕ, measured from south. These angles β and ϕ are expressed in the terms of the latitude of the place l, solar declination d, and hour angle h.

$$sin\beta = cosl.cosd.cosh + sinl.sind \tag{2.8}$$

and,

$$cos\phi = \frac{sin\beta.sinl - sind}{cos\beta.cosl} \tag{2.9}$$

The hour angle (in degree) is calculated as :

$$h = (12 - LST) \times 15 \tag{2.10}$$

If a surface is tilted by an angle Σ to the horizontal plane, then the incidence angle θ is given by

$$cos\theta = cos\beta.cos\gamma.sin\Sigma + sin\beta.cos\Sigma \tag{2.11}$$

When the surface is horizontal, $\Sigma = 0^o$, and :

$$cos\theta_H = sin\beta \tag{2.12}$$

For a vertical surface, $\Sigma = 90^o$, and :

$$cos\theta_V = cos\beta.cos\gamma \tag{2.13}$$

The surface solar azimuth angle γ is given as :

$$\gamma = \phi - \psi \quad \text{, before solar noon} \tag{2.14}$$

$$\gamma = \phi + \psi \quad \text{, after solar noon} \tag{2.15}$$

where, ψ is surface azimuth angle which represents the orientation of the surface. The ψ values for different orientation are $\psi_S(0^o)$, $\psi_E(90^o)$, $\psi_N(180^o)$ and $\psi_W(270^o)$.

2.3.5 Problem Formulation

2.3.5.1 Heat Transfer through Walls and Roof

Heat Transmission through the wall or roof of building structures is not steady and is therefore, difficult to evaluate. The two principal factors causing this are:

(a) The variation in the outside air temperature over a period of 24 hours.

(b) The variation in the solar radiation intensity that is incident upon the surface over a period of 24 hours.

The phenomenon is further complicated by the fact that a wall or roof has a thermal capacity due to which a certain amount of heat passing through it is stored and is transmitted to outside or inside at some later time.

The problem requires a solution of the governing equation for unsteady-state one-dimensional heat transfer, viz.,

$$\frac{\partial t}{\partial \tau} = \alpha \times \frac{\partial^2 t}{\partial x^2} \tag{2.16}$$

where, t is the temperature at any section of the wall or roof at a distance x from the surface at a time τ, and α is the thermal diffusivity given by

$$\alpha = \frac{k}{\rho.c} \tag{2.17}$$

where, k is the thermal conductivity and $\rho.c$ is the heat capacity of the wall or roof, in which ρ and c are the density and specific heat, respectively.

The heat transfer equation is to be solved with the boundary conditions of periodic variation of outside air temperature and solar radiation.

2.3.6 Sol-air Temperature

The calculation of structural load as a result of heat gain through exterior roof and walls involves the concept of sol-air temperature [6]. A heat balance at a sunlit surface gives the heat flux to the surface as:

$$q = \alpha_s I_t + h_o(T_o - T_{so}) - \epsilon \Delta R \tag{2.18}$$

ϵ = emissivity of the surface, 1.0 for black body

ΔR = 63.0 W/m^2, for horizontal surface and 0, for vertical surface [5]

α_s = absorptivity of the surface Equation 2.17 can also be written as

$$q = h_o(T_{sol} - T_{so}) \tag{2.19}$$

then, from equations 2.17 and 2.18 :

$T_{sol} = T_o + \alpha_s\ I_t/h_o$ - $\epsilon\ \Delta R/h_o$

Thus, the combined effect of the solar radiation and outside air temperature has been incorporated into a single effective temperature known as *sol-air temperature.*

2.3.7 Heat Transfer Coefficients for Walls and Roofs

The outside film coefficients are given by [8,16]:

For exterior walls

$$h_o = 7.373 + 5.066 \times V_o \quad , (W/m^2.^oC) \qquad (2.20)$$

For roof

$$h_o = 7.9953 + 6.364 \times V_o \quad , (W/m^2.^oC) \qquad (2.21)$$

where, V_o is the outside air velocity in m/s.

The inside film coefficients are given by [8]:

For interior walls

$$h_i = 1.77 \times (\Delta t)^{0.25} + (7.373 + 5.066 \times V_i) \quad , (W/m^2.^oC) \qquad (2.22)$$

For roof

$$h_i = 1.31 \times (\Delta t)^{0.25} + (7.9953 + 6.364 \times V_i) \quad , (W/m^2.^oC) \qquad (2.23)$$

where,
$\Delta t = |t_{si} - t_i|$,
t_{si} = inside surface temperature,
t_i = inside temperature, and
V_i = inside air velocity in m/s.

2.3.8 Heat Transfer Through Walls And Roofs

If the thermal capacity of the wall is ignored, then the instantenous rate of heat transfer through the wall at any time is given by

$$Q = UA(T_e - T_i) \qquad (2.24)$$

However, most building materials have a finite thermal capacity C which is expressed as

$$C = mc = \rho c V = \rho c (A \Delta X) \qquad (2.25)$$

where,
m = Mass of wall, [kg]
ρ = Density of wall material,[m^3/kg]

A = cross sectional area of wall material,[m^2] and
ΔX = wall thickness [m].

It has been seen that there is a two-fold effect of the thermal capacity on heat transfer

- a time lag between the heat transfer at the outside surface and at the inside surface
- decrement in heat transfer due to the absorption of heat by the wall and subsequent transfer of a part of this heat back to the outside air when the temperature of outside air is lower [10].

when these facts are taken into account, the actual heat transfer through the structure at any instant 't' is given by the following relation[2.26]

$$\dot{Q}_t = \sum_{j=1}^{5}[A_j U_j (T_e - T_i) + A_j U_j \lambda_j (T_o - T_e)] \tag{2.26}$$

where,
T_e = mean sol air temperature ^{o}C,
λ = decrement factor,
T_o = temperature at time ø before the time 't',
o = time-lag (h). $j = \mathrm{j}^{th}$ surface.

If thickness Δx is less than 0.30m then

$$\lambda = 1 - \frac{0.8 \times \Delta x}{0.30} \tag{2.27}$$

or$\lambda = 0.1$

U, A,and T_o refer to the overall heat transfer coefficient, cross sectional area and temperature at time $t - \phi$, respectively. The summation symbol $\sum$ stands for the four walls and the roof.

When the wall is very thin, the time lag factor $\lambda_j \to 1$ and we get

$$\dot{Q}_t = \sum_{j=1}^{5} A_j U_j \lambda_j (T_o - T_i) \tag{2.28}$$

On the other hand, in case of very thick wall $\lambda_j \rightarrow 0$ and the heat transfer is obtained from,

$$\dot{Q}_t = \sum_{j=1}^{5} A_j U_j (T_e - T_i) \tag{2.29}$$

The overall heat transfer coefficient U is given by

$$\frac{1}{U} = \frac{1}{h_o} + \sum_{j=1}^{3} \frac{\Delta x_j}{k_j} + \frac{1}{h_i} \tag{2.30}$$

T_o = ambient temperature(^{o}C) T_i = inside temperature(^{o}C) Δx_j and k_j = thickness and thermal conductivity of the 'jth' layer of the structural material h_i = convective heat transfer co-efficient for inside surface.kW$/m^2.^o$C U = overall heat transfer co-efficient, kW$/m^2.^o$C Figures 2.6 and 2.7 give the structural detail of walls and roof, respectively.

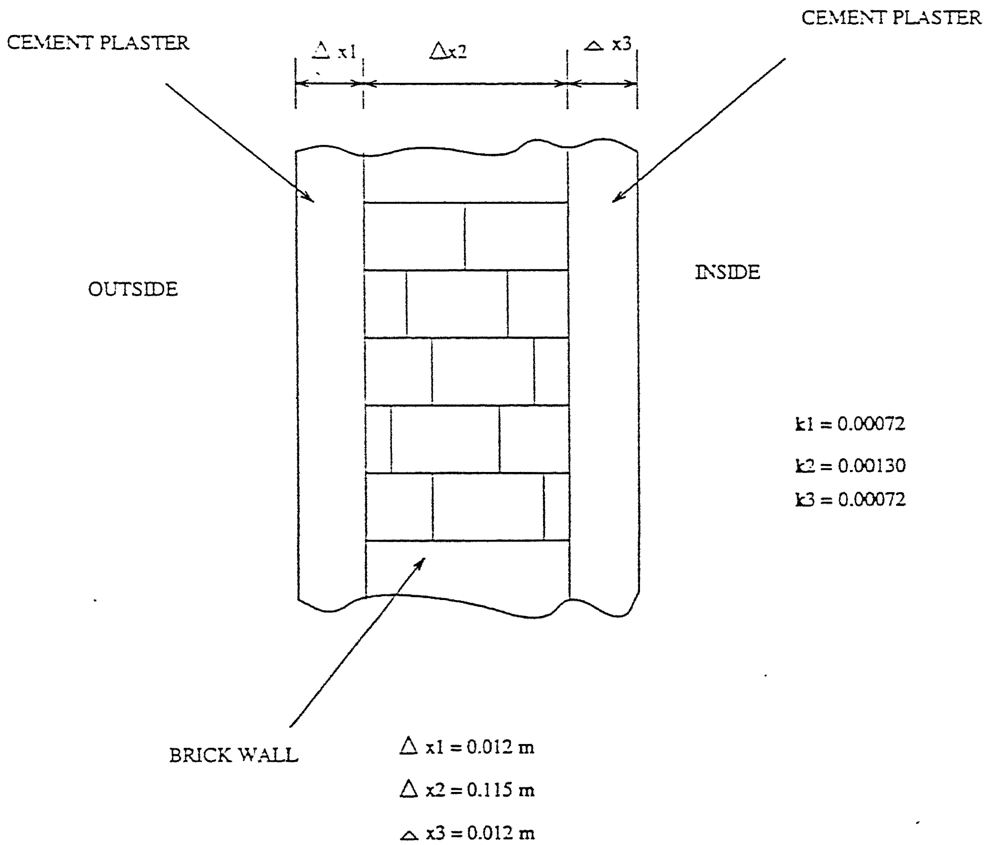

Figure 2.6: Structural detail of wall.

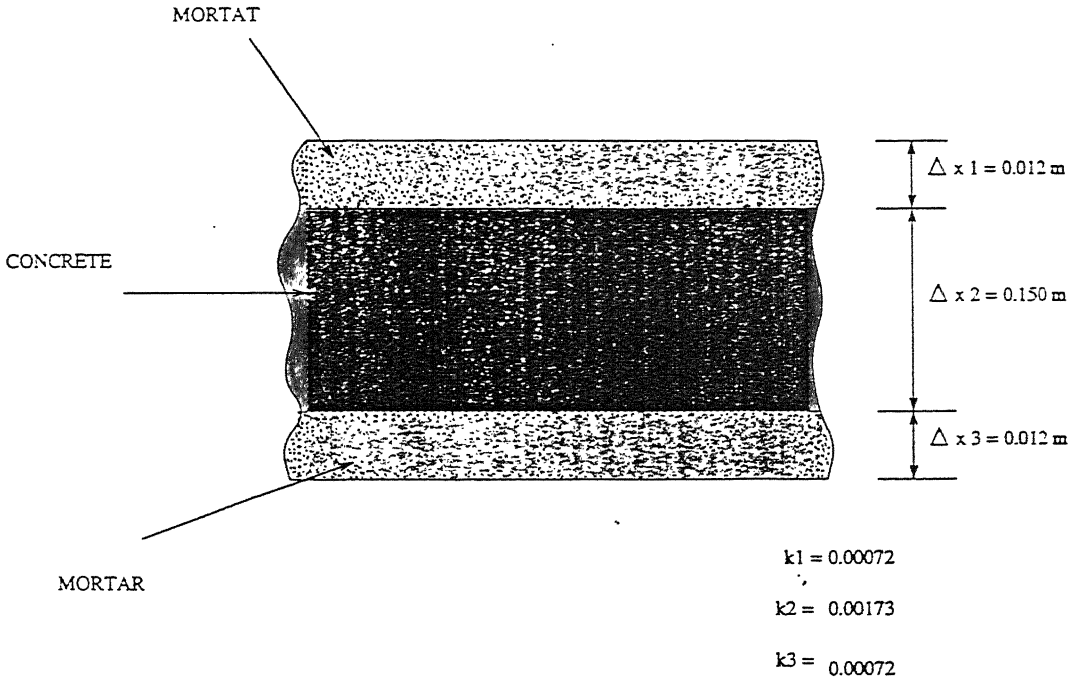

Figure 2.7: Structural detail of ceiling.

2.3.9 Heat Transfer through Glass

Glass construction forms a significant part of modern building structures. The heat transfer through glass [10] comprises:

- all the transmitted radiation
- a part of the absorbed radiation that enters the conditioned space, and
- the heat transmitted due to temperature difference between the outside and inside temperatures.

The direct radiation enters the space only if the glass receives the direct rays of the sun. The diffuse radiation enters the space even when the glass is not facing the sun. Figure 2.8 represents all sorts of heat transfer into conditioned space through glass. The heat transfer to the space is

give by :

$$\dot{Q}_{glass} = A_{sun}.\tau_D.I_D + A_{win}.\tau_d.I_d + h_i..A_{win}.(t_{gi} - t_i) \tag{2.31}$$

where,
t_{gi} = temperature of inner surface of glass
A_{sun} = glass area directly exposed to the sun
A_{win} = total glass area of window
The subscripts D and d denote the terms for direct and diffuse radiations, respectively.

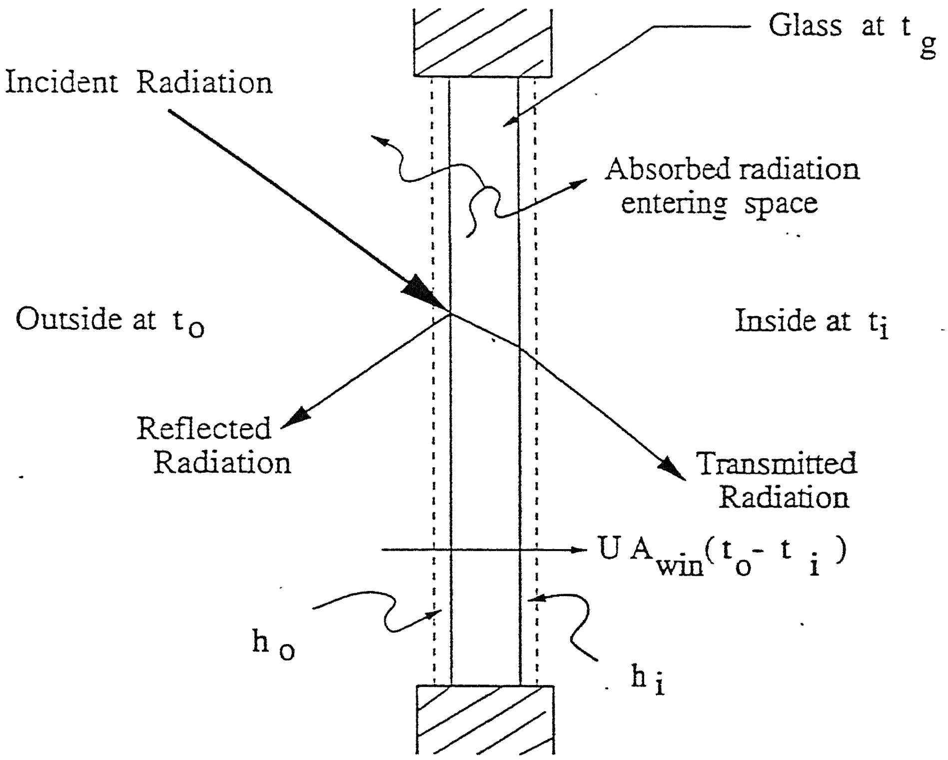

Figure 2.8: Heat transfer through glass.

Writing energy balance for glass sheet itself, we have

$$A_{sun}.\alpha_D.I_D + A_{win}.\alpha_d.I_d = A_{win}[hi.(t_{gi} - t_i) + h_o.(t_{go} - t_o)] \tag{2.32}$$

where, t_{go}=temperature of outer surface glass

As the thickness of glass sheet is very small, inner and outer surface temperatures of glass can be assumed to be equal to (t_g), without creating appreciable error in cooling load but simultaneously simplifying the analysis tremendously. Now, modified equation can be expressed as :

$$t_g = \frac{A_{sun}.\alpha_D.I_D + A_{win}.\alpha_d.I_d + A_{win}.h_i.t_i + A_{win}.h_o.t_o}{A_{win}(h_i + h_o)} \tag{2.33}$$

Putting the value of t_g in place of t_{gi} in equation (2.45) ,we get the cooling load due to glass as :

$$\dot{Q}_{glass} = A_{sun}.\tau_D.I_D + A_{win}.\tau_d.I_d + \frac{A_{sun}.\alpha_D.I_D + A_{win}.\alpha_d.I_d}{\left(1. + \frac{h_o}{h_i}\right)} + U..A_{win}.(t_o - t_i) \tag{2.34}$$

where, U is the overall coefficient of heat transfer given by

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_i} \tag{2.35}$$

If the thermal resistance of glass is also considered, U is given by :

$$\frac{1}{U} = \frac{1}{h_o} + \frac{\Delta x}{k_g} + \frac{1}{h_i} \tag{2.36}$$

where, Δx is the thickness of glass and k_g is its thermal conductivity.

Here, it has to be kept in mind that the transmissivity and absorptivity are function of the angle of incidence. They vary for direct radiation while are almost constant for diffused part. These values are given in Appendix 'D'[13].

2.3.10 Shading of Surfaces

The most effective way to reduce the solar load on fenestration is to intercept direct radiation from the sun before it reaches the glass. To serve this purpose, most glass areas are provided with reveals, overhangs and fins in the form of vertical and horizontal projections from the walls.

The ability of these projections to intercept the direct component of solar radiation depend on their geometry, surface-solar angle (γ) and the

profile or shadow-line angle (Ω) (Fig 2.9). The profile angle is defined as the angular difference between a horizontal plane and a plane tilted about a horizontal axis in the plane of the fenestration until it includes the sun. The profile angle can be calculated by:

$$tan\Omega = \frac{tan\beta}{cos\gamma} \tag{2.37}$$

The shadow height, S_H, due to horizontal projection, P, on a window

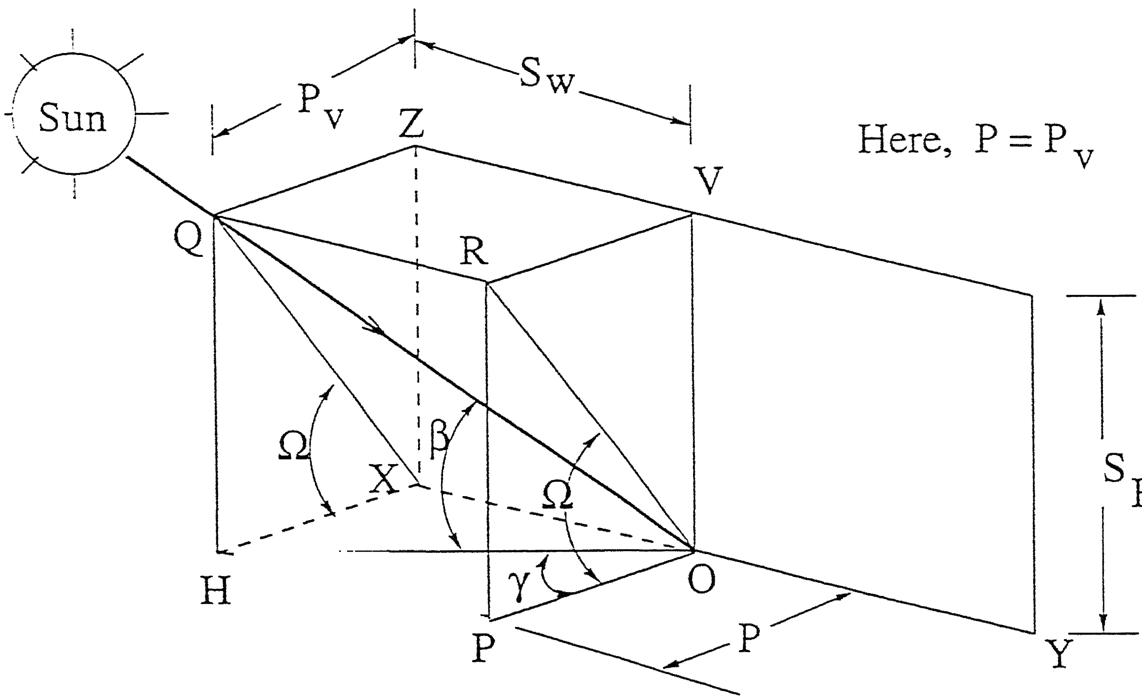

Figure 2.9: Shading on glass due to horizontal and vertical projections.

or wall for any time of day or year is related to the profile angle Ω by:

$$S_H = P \times cot\Omega \tag{2.38}$$

The shadow width S_w, due to a vertical projection, P_v, (it need not not be equal to the horizontal projection) on a window or wall for any given time of day and year is related to the vertical surface -solar azimuth angle γ by:

$$S_w = P_v \times cot\gamma \tag{2.39}$$

Thus, the sunlit area of the window is:

$$A_{sun} = (W - S_w).(H - S_H) \tag{2.40}$$

where, W and H are the width and height of window.

2.3.11 Heat Transfer through Door

The cooling heat gain due to doors are given by

$$\dot{Q}_{door} = U.A_{door}.(t_o - t_i) \tag{2.41}$$

where, U is the overall heat transfer coefficient through door [5] and A_{door} is area of the door.

2.4 Ventilation Load

Ventilation air is mandatory to ensure fresh air supply in the residential buildings. The amount of ventilation needed has been debated for over a century, and the different types of rationale developed have led to radically different ventilation standards. The current rationale [5] for the minimum outside air requirement is 2.5 L/s per person based on CO_2 concentration. The commonly accepted value is 2.364 L/s.

Ventilation, also, causes both types of load namely sensible heat gain ($\dot{Q}_{vs}$) and latent heat gain ($\dot{Q}_{vl}$). The load due to ventilation are given by following expressions :

$$\dot{Q}_{vs} = n \times 1.232 \times L \times \Delta T \quad , (W) \tag{2.42}$$

and,

$$\dot{Q}_{vl} = n \times 3012 \times L \times \Delta\omega \quad , (W) \tag{2.43}$$

where, ΔT is difference between outside and inside temperatures, ^oC. $\Delta\omega$ is difference between inside and outside air specific humidities, $kgw./kg\ d.a.$. L is ventilation air in litre per second per person and n is number of people in the conditioned space.

2.5 Occupancy Load

The occupants in a conditioned space give out heat at a metabolic rate that depends on their rate of working. The relative proportion of the sensible and latent heats given out, however, depends on the ambient dry bulb temperature. The lower the dry bulb temperature, the greater the heat given out as sensible heat. The instantaneous sensible load is the product of the sensible heat loss from the people and the cooling load factor (CLF). This CLF [5] is function of the time people spend in the conditioned space and the time elapsed since first entering. The sensible cooling load ($\dot{Q}_{os}$) and latent load ($\dot{Q}_{ol}$) can be expressed as :

$$\dot{Q}_{os} = n \times Sensible\ heat\ loss \times CLF \quad , (W) \tag{2.44}$$

and,

$$\dot{Q}_{ol} = n \times Latent\ heat\ loss \quad , (W) \tag{2.45}$$

where, n is number of people in the conditioned space.

2.6 Lighting Load

Electric lights generate a sensible heat equal to the electric power consumed. Most of the energy is generated as heat and the rest as light which also becomes heat after multiple reflections. An accurate estimate of the cooling load imposed by the lighting is not straight forward. Because the rate of heat gain to the air caused by lights can be quite different from the power supplied to the light points or lighting fixtures.

The time lag effect should be taken into account in calculating the cooling load, since the actual load is lower than the instantaneous heat gain, and peak load may be significantly affected.

The lighting load ($\dot{Q}_{ls}$) is expressed as :

$$\dot{Q}_{ls} = Light\ Wattage \times Special\ Allowance\ Factor \times CLF \tag{2.46}$$

where, cooling load factor (CLF) [5] is a function of time of use , type of arrangement, room furnishing, etc. Special allowance factor is introduced for fluorescent fixtures and fixtures requiring more energy than their rated Wattage. This is due to the fact that the choke takes about 20 - 25% of the rated power of the tube, used in mercury vapour tube lights.

2.7 Power Equipment Load

Power equipment load ($\dot{Q}_{ps}$) is calculated from

$$\dot{Q}_{ps} = \frac{kW\ Rating \times Load\ Factor \times 1000 \times CLF}{\%Motor\ Efficiency/100}\ , (W) \qquad (2.47)$$

where, *load factor* is merely the fraction rated load delivered under the conditions of the cooling load estimate.

Cooling load factor (CLF) [5] is the function of time (hours after equipments are on) and total operational time.

2.8 Appliances Load

Most appliances contribute sensible and latent heats. The latent heat produced depends on the functions the appliances perform. Electric motors contribute sensible heat to the conditioned space. A part of power input is directly converted into heat due to the inefficiency of the motor and is dissipated through the frame of the motor. The motor efficiency of 80% is considered in the case study presented. In estimating cooling load, heat gain from heat producing appliances , e.g. kitchen appliances, computers, etc., must be taken into account. Some appliances produce only sensible heat load while others generate sensible ($\dot{Q}_{as}$) as well as latent load ($\dot{Q}_{al}$).

$$\dot{Q}_{as} = Sensible\ Heat\ Rate \times CLF \qquad (2.48)$$

and,

$$\dot{Q}_{al} = Latent\ Heat\ Rate \qquad (2.49)$$

Here, it can be noticed that sensible load is present even after putting off the appliance while latent heat load is present only for that hours in which appliance is on.

2.9 Total Cooling Load

The solar load ($\dot{Q}_{ss}$) is the summation of

$$\dot{Q}_{ss} = \dot{Q}_{structural} + \dot{Q}_{glass} + \dot{Q}_{door} \tag{2.50}$$

Total cooling load is sum of all the sensible loads and latent loads.

$$\dot{Q}_{sensible} = \dot{Q}_{ss} + \dot{Q}_{vs} + \dot{Q}_{os} + \dot{Q}_{ls} + \dot{Q}_{ps} + \dot{Q}_{as} \tag{2.51}$$

and,

$$\dot{Q}_{latent} = \dot{Q}_{vl} + \dot{Q}_{al} + \dot{Q}_{ol} \tag{2.52}$$

Therefore,

$$\dot{Q}_{total} = \dot{Q}_{sensible} + \dot{Q}_{latent} \tag{2.53}$$

The design cooling heat load is $S.F. \times \dot{Q}_{total}$ where,
$S.F.$ = safety factor =about 1.1.

2.10 Ventilation Air Calculations

The mass flow rates of ventilation air is calculated from the total cooling load $\dot{Q}$ for the conditioned space. Thus the evaporatively cooled ventilation air requirement for a temperature difference of ΔT is calculated by [16] in the following expressions:

$$\dot{Q} = \dot{m}_a \times c_p \times \Delta T \quad ,(kW) \tag{2.54}$$

$$\dot{m}_a = \frac{\dot{Q}}{c_p \times \Delta T} \quad ,(kg/s) \tag{2.55}$$

$$\dot{V} = \dot{m}_a v \quad ,(m^3/h) \tag{2.56}$$

where,

$\dot{m}_a$=mass flow rate of ventilation air requirement,(kg/s)

$\dot{V}$=Volume flow rate of ventilation air required , (m^3/hi)

Chapter 3

Evaporative Airconditioning And Duct Design

Air cooling by water evaporation occurs in nature near waterfalls, flowing air streams over lakes and oceans, under summer showers and even upon wetted skin. The evaporative process simply removes sensible heat (i.e., cooling by decreasing the surface temperature) and replaces it with latent heat (i.e., increasing the moisture content of air). It is interesting to note that the sea acts as an evaporative cooler due to which the temperature in costal areas of India hardly goes beyond 30 - 35°C as compared to 40 - 45°C in the hot-dry regions. This is supported from the detailed results presented in [18].

3.1 Theory

Evaporation is described as an adiabatic process, during which the total thermal energy contents of the system remains constant. Figure 3.1 shows the depression in temperature resulting from latent heat gain of air during evaporation of water. As water evaporates, the sensible heat content of the system falls, while the latent heat content increases by an equal amount. In other words, dry-bulb temperature of air falls, but its moisture content rises. The limit of temperature reduction is up to the wet-bulb temperature of the air at the beginning of the process. Surface

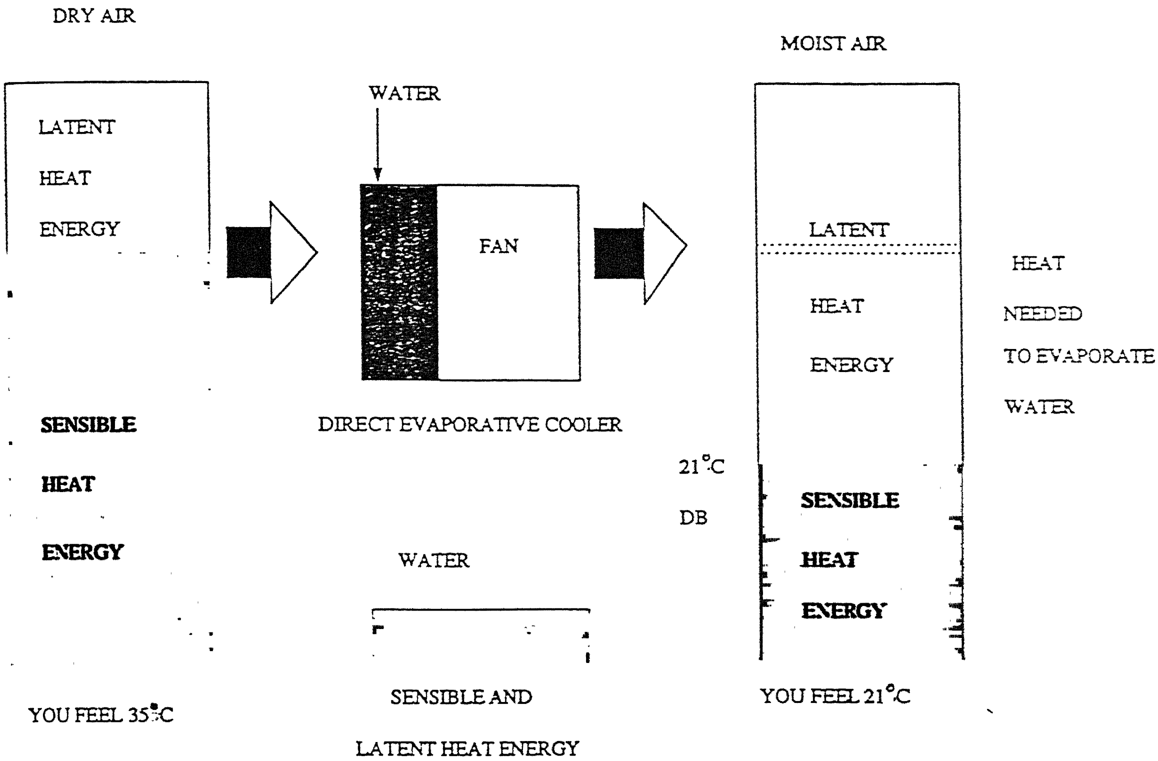

Figure 3.1: Schematic representation of depression in dry-bulb temperature during evaporative cooling [7].

evaporation under ideal conditions can cool up to the wet-bulb temperature. The process stops when the relative humidity of air approaches 100 %. A simple measure of the potential for evaporative cooling at any given air condition is the wet-bulb depression, defined as the difference between the dry-bulb and wet-bulb temperatures. It provides an upper limit of the achievable temperature drop.

Water is a universal coolant and the working fluid that can meet air-conditioning needs in both residential, commercial and industrial applications. Two principle methods of evaporative airconditioning are commonly used:

- *Direct Cooling Process*
- *Indirect Cooling Process*

Direct cooling in which water evaporates directly into the air stream,

thus reducing the air dry-bulb temperature, while humidifying the air.

Indirect cooling, where primary air is cooled sensibly with a heat exchanger, while the secondary air carries away the heat energy from the primary air as generated vapour. Direct and Indirect processes can be combined (indirect/direct). Compared to vapour compression systems, increased air flow rates are used for direct evaporative comfort cooling to compensate for higher supply air temperatures.

3.1.1 Direct Evaporative Airconditioning

In direct evaporative cooling air is drawn through wetted pads or a spray chamber and its sensible heat energy causes evaporation of water, the dry-bulb temperature of air gets reduced. The air temperature is decreased by 60% to 90% of the wet-bulb depression (ambient dry- bulb temperature less wet-bulb temperature.) Note that there is no sensible cooling and that this is essentially an isenthalpic process for direct evaporative airconditioning (also termed evaporative, "cooling").

In arid regions direct coolers provide comfort cooling, while high humid areas can use direct cooling for specialised applications. Direct EAC consumes significantly less energy than vapour-compression refrigeration. The only power consuming components of an evaporative cooler are fans and small water pumps. Energy savings of evaporative coolers vary with humidity levels and temperature. Direct systems in low-humidity zones typically realize an energy saving of 60% to 80% over refrigerated systems. In-direct evaporative cooling equipment, water is supplied through a float valve through a small reservoir from where it flows through fibrous pads. A fan draws large volume of outdoor air through the pads, where it is cooled by evaporation, and then supplied to the building. This cool and more humid air absorbs sensible heat from the buiding. An efficient wetted pad can reduce the air temperature by as much as 95% of the wet-bulb depression, while an inefficient and poorly designed pad may only reduce this by 50%, or less.

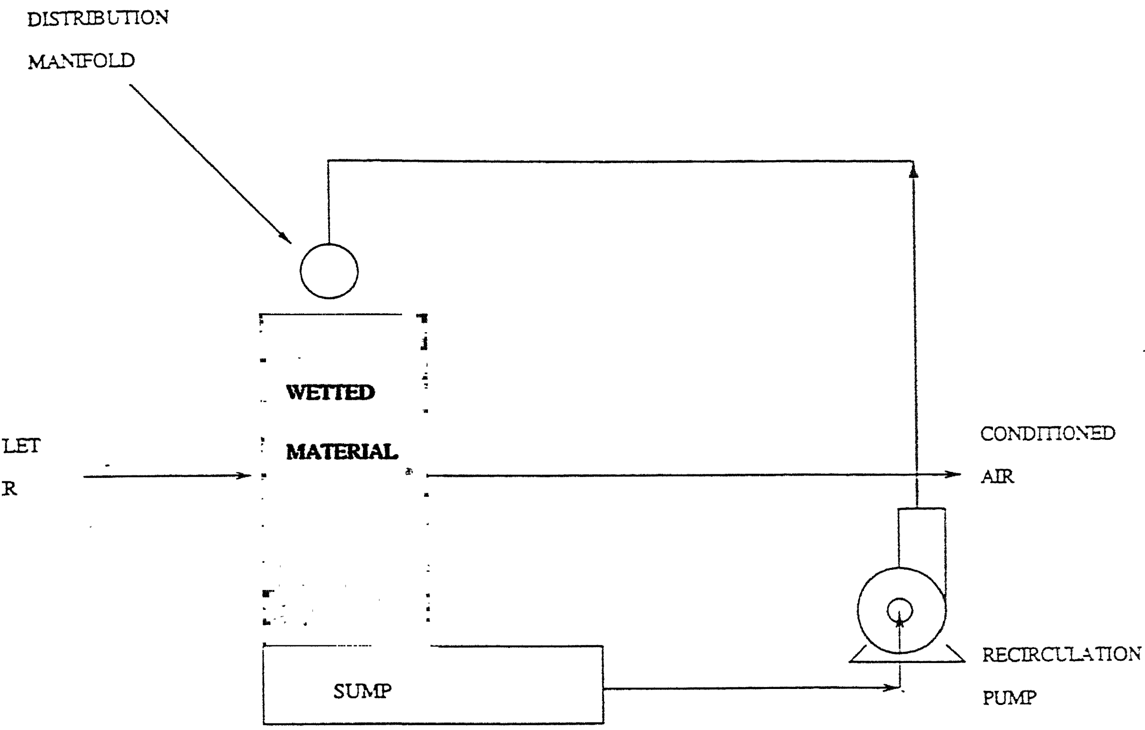

Figure 3.2: Schematic representation of direct evaporative cooling system.

In the actual evaporative cooling process there is very little change in the energy state of air. Direct ECA is simple and cheap due to the reduced energy requirement for fan and make up water, but it is ineffective if ambient wet-bulb temperature is close to dry-bulb temperature, the cooling effect is not sufficient enough for indoor comfort cooling applications, but it finds some applications (e.g.,green houses and industries spot cooling). Direct evaporative coolers should not recirculate indoor air as cooling effectiveness decreases.

The saturation effectiveness of a direct evaporative airconditioner best describes the performance of an ECA unit. Saturation effectiveness is described as the difference between the entering and exit dry-bulb temperatures over the wet-bulb depression and can be defined as [7]:

$$\eta_{se} = (T_{db} - T_{supply})/WB_{depression} \tag{3.1}$$

where,

η_{se}=saturation efficiency
$WB_{depression}$=$(T_{db}$ - $T_{wb})_{outside}$
T_{db}=outdoor dry-bulb temperature
T_{wb}=outdoor wet-bulb temperature

3.1.2 Indirect Evaporative Cooling

In indirect evaporative cooling water does not evaporate in the air supplied into the conditioned space. It attempts to make use of the evaporative cooling process without increasing the amount of moisture in the supplied air. Indirect evaporative cooling equipments use a heat exchanger to avoid the direct contact between water to be evaporated and the supply air. A direct evaporative process cools air that flows across one side of the heat exchanger, removing heat, and is then exhausted to the atmosphere. The air to be supplied to the building flows across the other side of the heat exchanger and is cooled without receiving any moisture (Fig. 3.3) The processes are:

Process 1 - 2 : Humidification of secondary air.

Process 2 - 3 : Heat from primary air transfered to secondary air.

Process 1 - 4 : Cooling of primary air for sopply to the conditioned room.

The second direct stage is then added to further cool the air. In this system the outside air is pre-cooled in an indirect stage and then further cooled in a subsequent direct stage. The first stage cools the air without adding moisture and in second stage air passes directly into the water spray system. This generally yields the final air temperature leaving the evaporative cooler about 3.5°C lower than what could be achieved with a direct ECA only. This expands the ECA considerably to areas with slightly higher wet-bulb temperatures. Generally, effectiveness of an indirect stage evaporative cooling system is found to be 65% is reached which allows ambient wet-bulb temperature of upto 25°C to provide low enough room temperature for real comfort cooling.

The performance of the indirect evaporative cooling is measured by PF

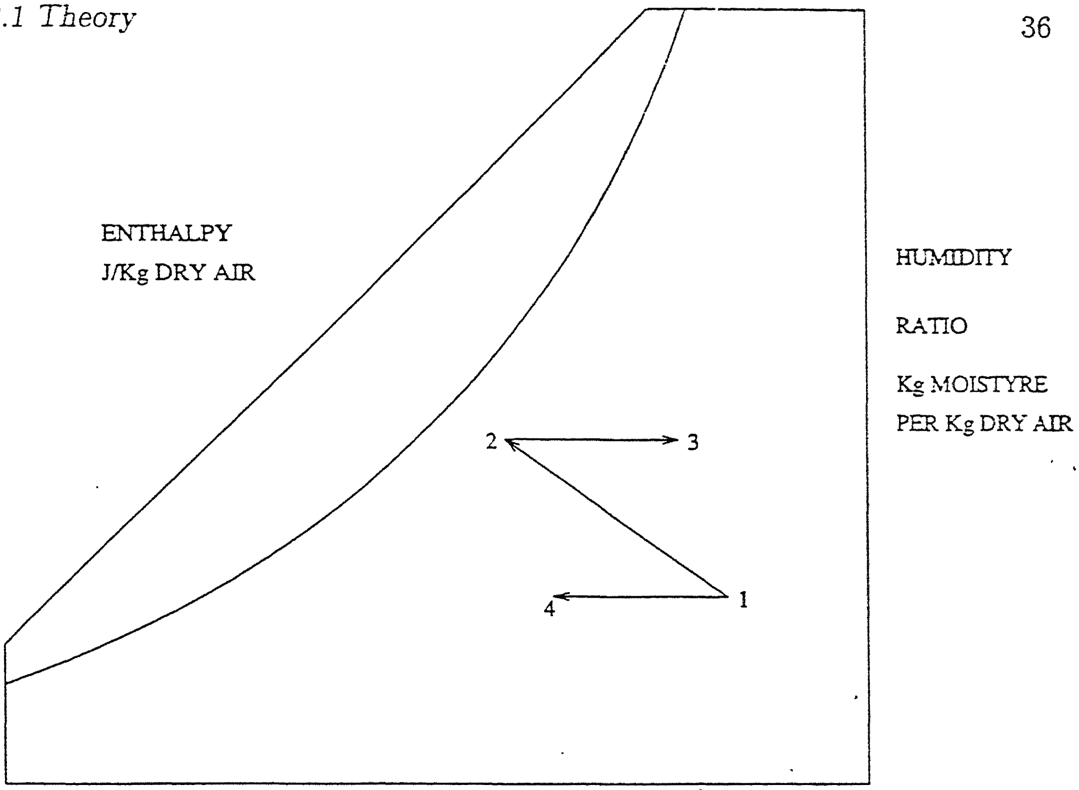

Figure 3.3: Schematic representation of different processes taking place during evaporative cooling

which is the ratio of the reduction of the dry-bulb temperature of the dry side air stream to the initial difference between the dry side dry-bulb and wet side wet-bulb temperatures [7].

$$PF = (T_{db} - T_{supply})/WB_{depression} \tag{3.2}$$

where,
PF=performance factorncy
$WB_{depression}$=(T_{db} - $T_{wb})_{outside}$
T_{db}=outdoor dry-bulb temperature
T_{supply}=supply air dry-bulb temperature

An indirect evaporative cooling process is illustrated in fig. 3.5 In this method the return air at a temperature lower than the ambient temperature is used to cool the fresh air in a surface-to-surface heat exchanger. The system as well as the processes are shown in figures 3.4 and 3.5

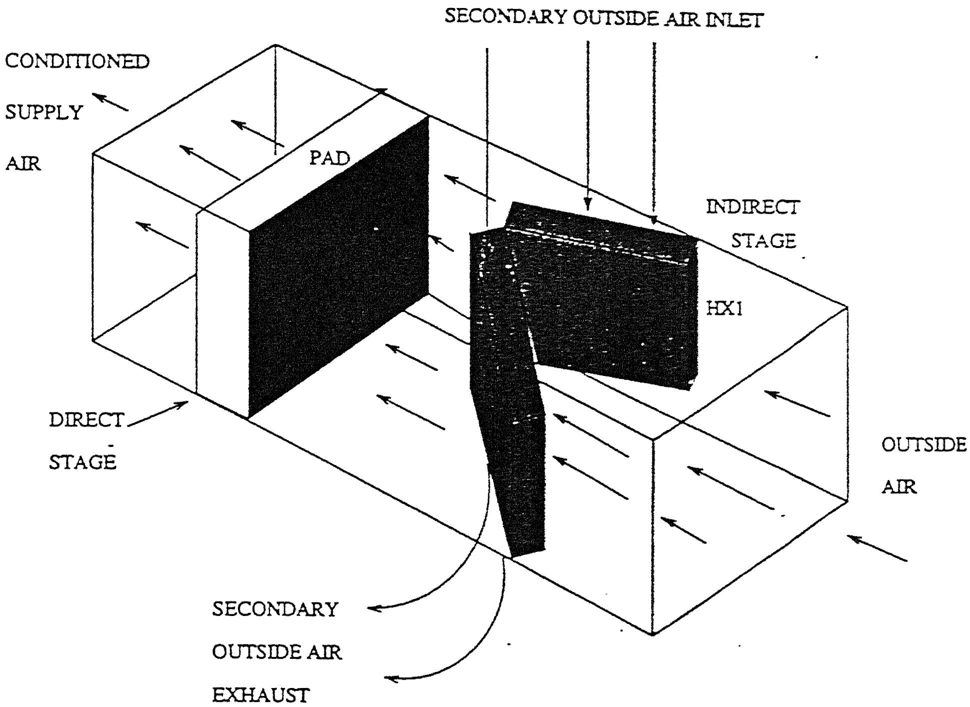

Figure 3.4: Schematic representation of plate type Indirect/Direct evaporative cooling airconditioner

respectively.

The processes are:

Process 3 - 4 : Humidification of primary air from the room.

Process 4 - 2 : Heat gained from the room by the humidified air

Process 1 - 3 : Sensible cooling of primary air by the secondary return air from the room.

Process 2 - 5 : Sensible heat gain by the room return air in the surface to surface heat exchanger HE1.

By this arrangement the temperature of the humidified air is further lowered by a few degrees. Hence,it provides better comfort. If it is desired to get an even lower temperature, the return air can be passed through a humidifier. By this approach the dry-bulb temperature of the

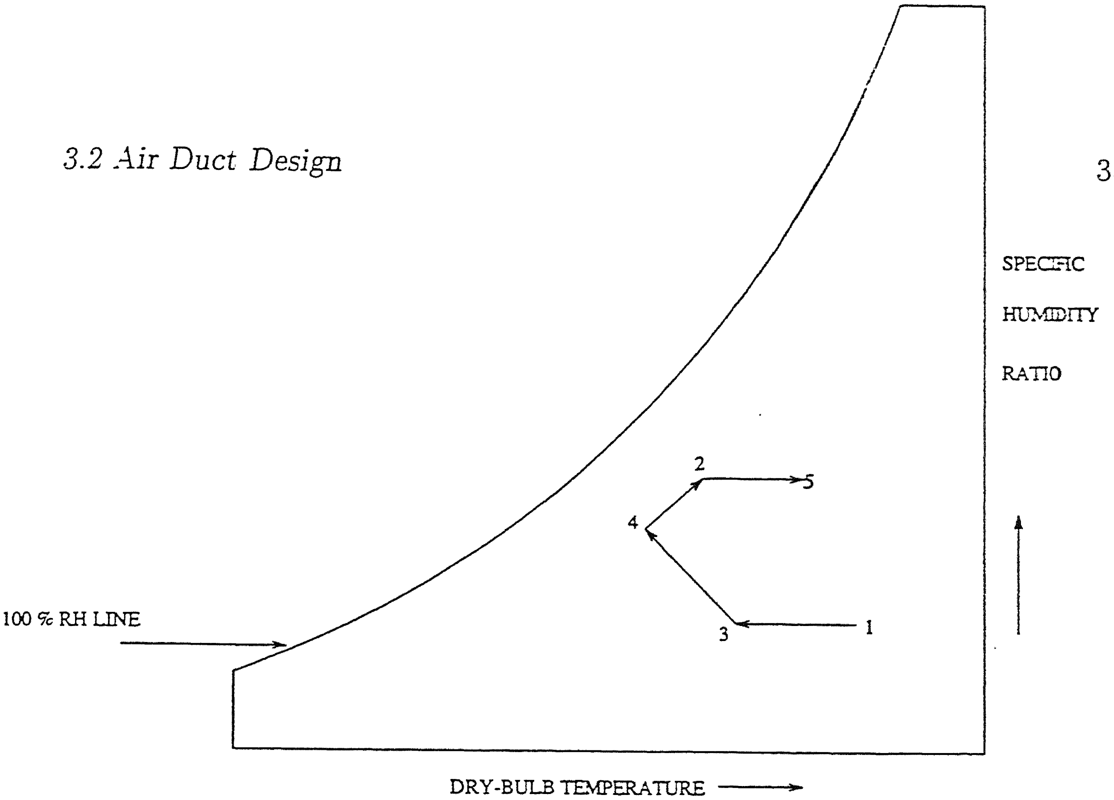

Figure 3.5: Schematic representation of indirect/direct evaporative cooling process

return air is further lowered. As such it renders further reduction in the dry-bulb temperature of the primary air passing the surface-to- surface heat exchanger. Thus after humidification of this cooled air renders temperature which may be very close to the mechanical airconditioning state. The processes for the latter modified system is represented by 1 - 3 - 4 - 2 - 6 - 7 in fig.3.6.

3.2 Air Duct Design

The essential economics of an air transmission system is achieved by a proper balance between the initial or final cost for the given flow rate of air. The first cost is detirmined by the cost of the duct system which depends on the duct size. The operating cost is detirmined by the fan power consumption which depends on the friction pressure drop in the evaporative cooling and duct system. The pressure drop can be reduced by increasing the size of the ducts but it will increase the first cost. Hence, there should be a proper balance between duct size and operating cost. A few general rules are stated below which may form as a guideline

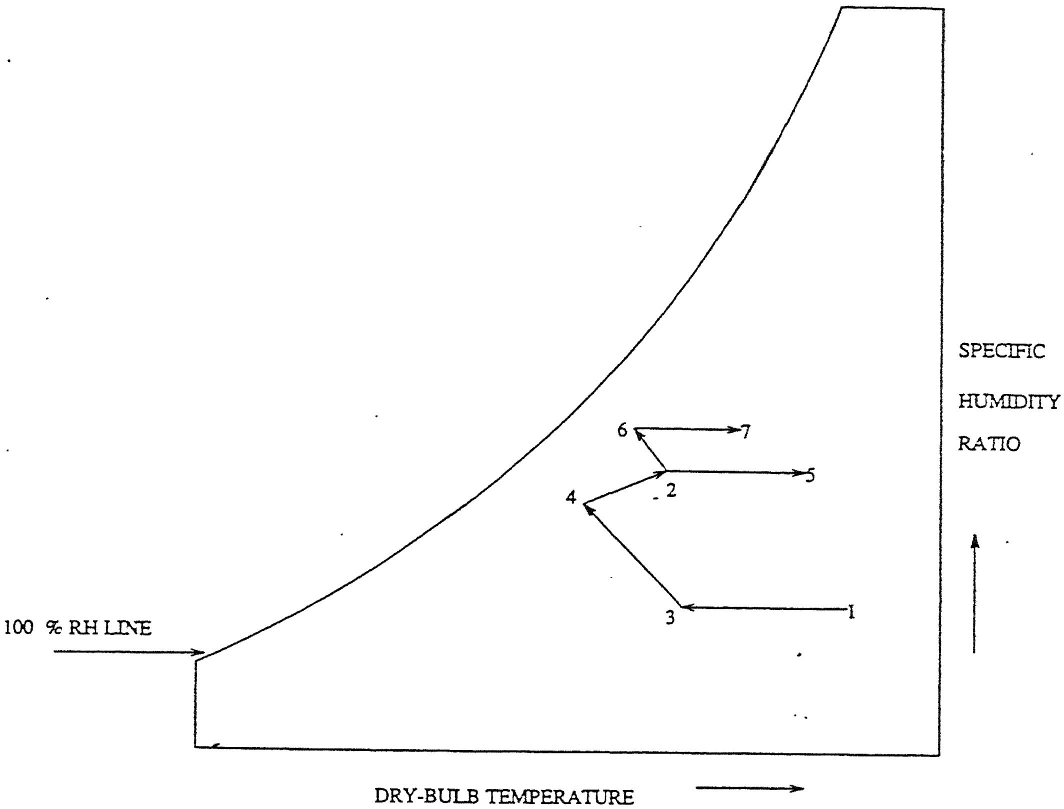

Figure 3.6: Schematic representation of an indirect/direct evaporative cooling process

for the design of ducts.

- Air should be conveyed as directly as possible to economize power, material and space.
- Sudden changes in direction should be avoided. When bends are essential, turning vanes should be used to minimize the pressure losses.
- Air velocities in ducts should be within permissible limits to keep the noise level within allowable limits.
- Diverging sections chould be made gradual. The angle of divergence should not exceed 20^o[12,13].
- Rectangular ducts should be made as nearly square as possible. However, keeping the dimensions as constant as possible also helps reduce the use of material and construstion cost. This will ensure

minimum duct surface, and hence cost for same air carrying capacity. An aspect ratio of 4:1 or less should be maintained.

- Ducts should be made of smooth material such as galvanised iron (GI) or aluminium sheet. Whenever other material is used allowance should be made for roughness of material.

3.2.1 Duct Design Procedure

There are three common methods for sizing of ducts. They are:

(a) Equal Friction Method : In the equal friction method, the frictional pressure drop per unit length of the duct is maintained constant throughout the duct system. The procedure is to select a suitable velocity in the main duct based on practical requirements. Thus by knowing the air flow rate and velocity in the main duct, the size and friction loss are detirmined from the frangers charts or by using the expression of frictional pressure drop. The frictional pressure drop is found from:

$$\frac{\Delta p}{\Delta L} = 0.0206V^{1.845}/d^{4.92} \tag{3.3}$$

where,
V=volume flow rate of air, (m^3/s)
d=equivalent duct diameter, (m)
Δp=pressure drop equivalent to water head, (mm)
This expression gives the relation among pressure drop, volume-flow rate and equivalent diameter of duct.
If the volume-flow rate is given, and assuming the allowed velocity of the duct, the diameter of the circular duct can be found from:

$$d = \sqrt{4 \times V/\pi \times v}, (m) \tag{3.4}$$

where,
v=velocity of air in the duct, (m/s)
For known V, and d, the pressure drop per unit length is calculated. Thereafter, for any volume flow rate and known pressure drop d is

calculated as:

$$d = ((\Delta p/\Delta L)/[0.0206V^{1.845}])^{1/4.92}, (m) \qquad (3.5)$$

This method of sizing ducts automatically reduces the air velocity in the direction of flow. The method is generally recomended because of simplicity. If an equal friction design has a mixture of short and long runs of ducts, the shortest duct will need considerable amount of dampering. This is a draw back of the equal friction design.

(b) Velocity Reduction Method: In this method the main duct is designed in the same manner as in the equal friction method. There after arbitary reductions are made in the air velocities as we go down the duct run. The equivalent diameters are found as in the first case from the friction chart. Though the method allows safe velocities, it is not normally addopted unless the person using it has considerable practical experience and knowledge to design within reasonable accuracy.

(c) Static Regain Method: The principle of the static regain method is to maintain a constant static pressure before each terminal and each branch. This is achieved by sizing the duct in such a manner that after each branch or outlet, the static pressure gain due to the reduction in velocity exactly balances the pressure drop in the succeeding duct design section[12,13].

3.3 Rectangular Equivalent of Circular Ducts

Air ducts are usually sized first for round sections. Then if rectangular ducts are required, ducts are designed to provide the same flow rates and to have the same rate of pressure drop as for round ducts. The rectangular equivalent of a round duct is given by[10,12,13] as:

$$D_e = 1.302[\frac{(ab)^5}{(a+b)^2}]^{1/8}, (m) \qquad (3.6)$$

where,
D_e=equivalent diameter of the circular duct,(m)
a and b=sides of the rectangular duct,(m)

The equation may be used to detirmine one dimension of a rectangular duct if the other is assumed equivalent to a circular duct whose diameter is known. Although round ducts require the least metal to carry a given quantity of air, rectangular ducts are used often because of the following reasons:

- Space considerations as they fit easily in the building construction and occupy less building space without being conspicuous.
- Ease of fabrication.

Square ducts are closest to round ducts. They require less material than rectangular ducts. The material required for the rectangular ducts increase with the aspect ratio, viz a/b.

Chapter 4

Duct Design Procedure and Experimental Setup

4.1 Duct Design and Layout

The steps for calculation of the ventilation air are given in chapter 3 sec. 3.6.for different volumes of ventilation air for each floor of the triple storey building, and the following procedure was adopted for the ducts:

- Knowing the total volume of air required per floor, firstly air distribution for the different parts of the building was planned on the basis of the users requirements.
- The ducts layout was planned keeping the following as guidelines:
  - The space restrictions within the building.
  - Equal distribution of air for comfort of all occupants.
  - The users requirements.
  - Balanced system.
- The friction charts of air were used to evaluate the equivalent diameter of the ducts keeping a constant pressure drop per length of duct and the known volume flow rate of the air as planned earlier.
- A computer program was used to convert the equivalent diameter to a rectangular section using the expression given in chapter 3. The

conversion charts were also used for this purpose. Care was taken to provide a nearly square section of the rectangular duct to incorporate maximum saving of construction material but, at the same time due to space restrictions within the structure a compromise betweenw the two had to be resorted to.

- All the bends were provided with curved vanes within the duct to reduce pressure losses within the ducts.
- The air velocity in the main ducts were taken to be 9 m/s, being the allowable velocity for tolerable sound level for industrial applications. Though the duct layout was found to be better and economical when the layout was as per the drawings attached. These figures were modified in order to satisfy the users due to the machinery layout and lighting fixtures. These constraints cropped up after the design and drawing were completed. The modified air distribution system ditributes air in every two bays and satisfies the users need.

4.2 Choice of Axial-Flow Fan

The catelogues of standard manufactures were used to choose the axial-flow fans. The following were incorporated while making the choice

- Choice of low rpm fans to reduce noise levels.
- Small diameter fans to facilate the space restrictions.
- Delivery of the volume flow rate required per floor.
- Low electric energy consumption.
- Delivery of the required pressure head per floor. However in the present case the fan diameter is large for the required air delivery. The fan is a little costlier and the same is compensated by the reduced running bill.

4.3 Design of Evaporative Cooler System

Based on various guidelines of ASHRAE Book of Equipment[20] for the range of parameters were designed for evaporative cooling systems. No strict norms are available, and the system design was made keeping the latest developments in mind. Generally straight evaporative pads are most commonly used having the water distribution at the top. The water distribution is done through the holes in the pipes. This type of system is followed for evaporative cooling systems installed in India. The experiments to arrive at few results which were needed were performed as explained subsequently. A wood wool pad used as an eliminator is of 30mm thickness and 0.9 kg/m^2 of the eliminator surface area. This is to prevent any water carry over to the supply duct. This becomes essential as in the existing evaporative cooling supply duct the carry over of water droplets to the supply duct and finally its leakage through the duct was a serious problem. This point is taken care of by introducing the eliminator. The eliminator thickness is reduced in order to reduce the pressure drop in the air supply.

4.3.1 Experimental Setup

A pump was used to circulate water from the collection tray through a calibrated rotameter to a trough of dimensions $1000 \times 150 \times 50$. The trough was used to collect the water from the rotameter which was through a control valve. The water in the trough was then made to flow back to the collection tray through a number of holes of equal diameter drilled in the base of the trough.There were two rows of holes drilled in the trough 30mm apart. The distance between the holes in each row was kept at 60mm. The experiment was conducted for three set of hole diameter 2mm, 2.5mm, and 3.175mm respectively.The photographs of the experimental setup are attached below

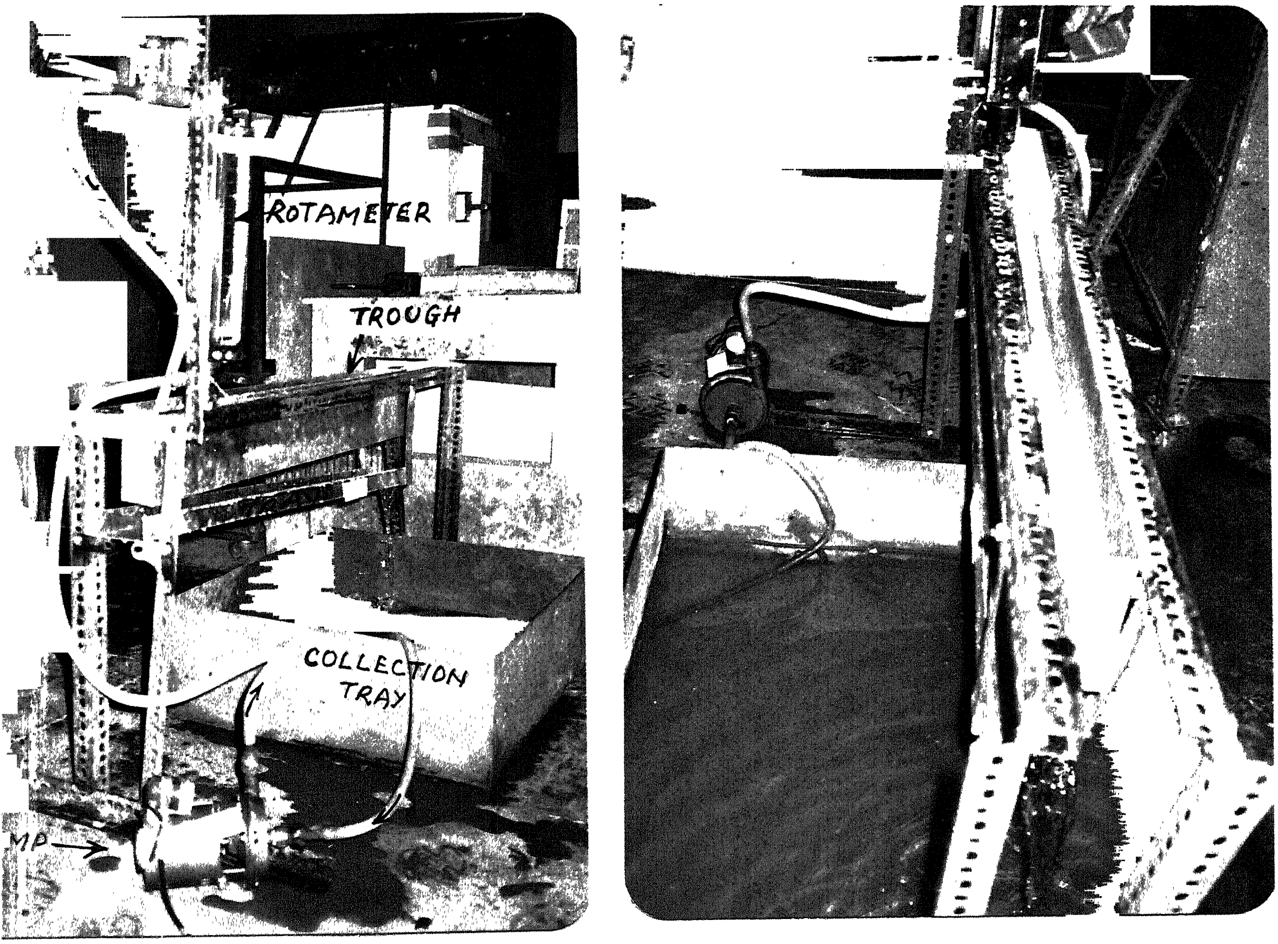

Figure 4.1: Experimental Setup

4.3.2 Experiment Details

- Water was made to flow through the rotameter at different flow rates controlled by the adjustable valve.
- For the different flow rates when the head of water in the trough became constant, the head and flow rates were measured and noted. This procedure was repeated for 2mm, 2.5mm, and 3.175mm diameter holes.

In this manner a set of readings were noted for the three different diameter hole sizes and a genral idea of how the Cd and head vary with the discharge.

4.3.3 Density of Pads

Since there are no laid down parameters for the density of wood wool to be kept per pad. A rough estimate given by ASHRAE is 1.5kg/m^2 to 2kg/m^2 of surface area for a 50mm thick wood wool pad.

In order to establishe the most appropriate pad density for the cooling system recomended a frame of $30cm \times 30cm \times 5cm$ was prepared and varying densities of wood wool fibres were packed into the frame to choose the most appropriate density by judgement. The photographs of the same are attached for reference

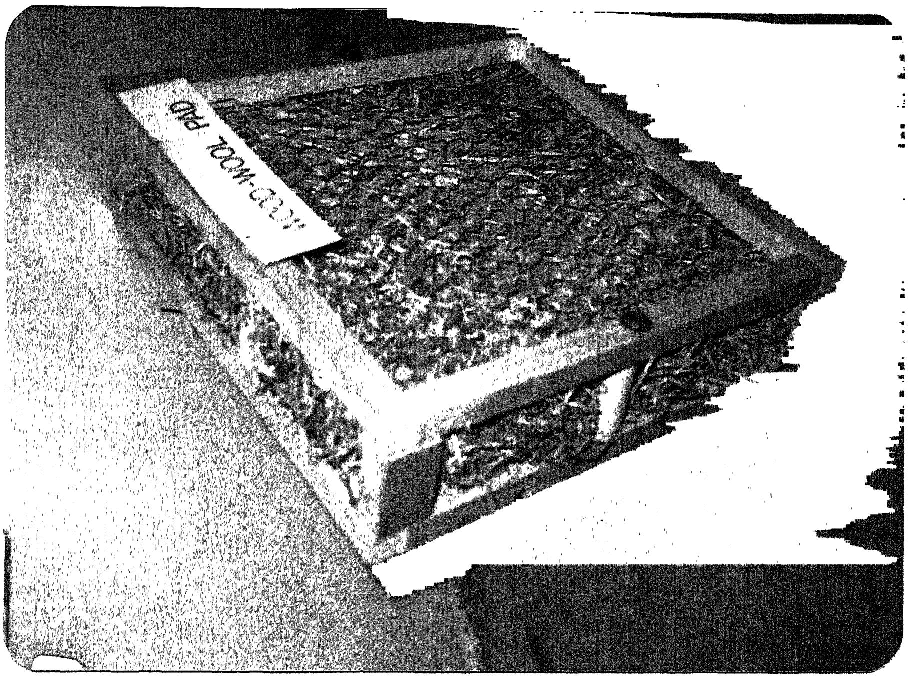

Figure 4.2: Wood Wool Pad 1.5kg/m^2 Density

Chapter 5

Results and Discussion

Calculations for the present case study carried out for the manufacturing shop at Kanpur enable an experience as to how far there is difference between the general approach and the pratical situation. This chapter deals with results obtained by calculation of cooling loads on the basis of the theoretical and actual temperature variation of outside air. For this a generalised computer programme[ ] is modified in order to in corporate formulation for detirmination of ventilation air supply for different floors fo the triple storey building.

5.1 Input Data

the input data for the present problem are : building geometry dimensions, orientation, location, machinery details, light fixtures, number of occupants etc. Hence, the total cooling load was found for different days and months.

For calculation of the ventilation air, maximum of maxima found for different months was finally identified for which the ventilation air was detirmined.

5.2 Graphical Representation of Results

5.2.1 Theoretical and Actual Temperature Variation

The actual hourly temperature variation as well as the suggested hourly variation in temperature based on maximum and minimum temperature variation of the day have been shown in figures 2.1 to 2.4. The theoretical temperature generally matches very well with the actual values except deviating here and there with 2 - 3 K. The deviation between them are such that over prediction at one time gets nullified by under prediction at other time. Moreover the present approach simplifies calculation procedure significantly.

5.2.2 Variation of Wet-Bulb Temperature

Figure 5.1 shows the variation in wet-bulb temperature for the month of April, May, June and July for Kanpur city. There is not much variation in WBT in a particular day. The maximum WBT occurs at about 3 P.M. except in July, in which the maxima occurs at about 10 A.M. This variation may be different for different places.

5.2.3 Variation of Relative Humidity

Figure 5.2 shows variation in relative humidity with time for the month of April, May, June and July for Kanpur city. The relative humidity for these months is minimum at about 3 P.M. when dry-bulb temperature is maximum.

5.2.4 Cooling Load Variations

The total cooling load on the structure varies during the day as per magnitudes of the maximum and minimum outside temperatures. The variation of the total cooling load is represented by graph for each floor

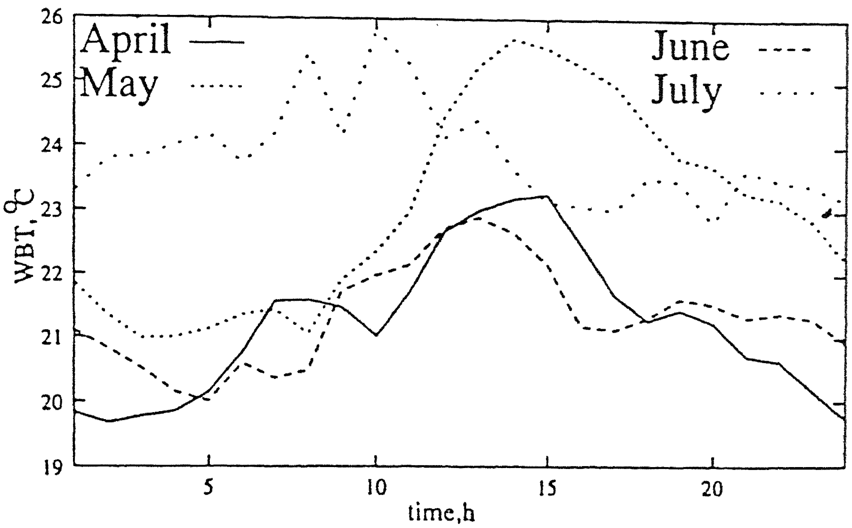

Figure 5.1: Hourly variation of wet-bulb temperature.

of the production shop as shown in figure 5.3. The maximum cooling load on the building occurs around 5 P.M., i.e.. about 5 hours after the peak solar load is felt outside. This is due to the heat capacity of the building and the various openings in the building. Figure 5.3 show the effect of the decrement factor and the time lag on the cooling load. It means that a building's material and thickness of wall play a significant role in the solar heat gain by the building. It is also insteristing to note that though the instantaneous solar heat gain is very high, the wall thermal capacity helps reduce the peak cooling load, i.e., the lower capacity refrigeration equipment may be used to take care of cooling of the building. Hence, the initial investment gets significantly reduced. But the same may not render significant reduction in the running cost, because the heat retained by the wall during the solar heat gain is reduced during the night, causing increased cooling compared to the existing lower thermal potential during the evening or night.

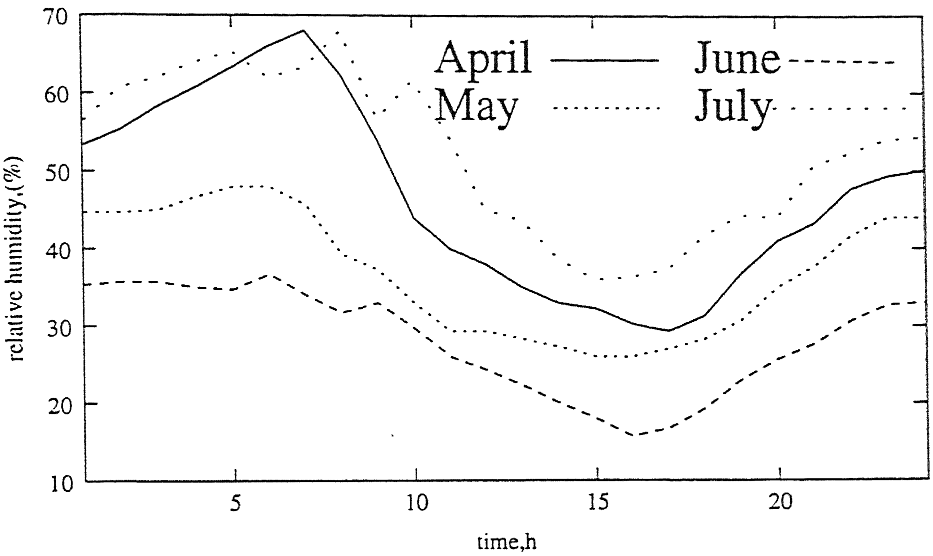

Figure 5.2: Hourly variation of relative humidity.

5.2.5 Experimental Results

The graphs are plotted to show how the discharge coefficient varies for different hole diameters. A graph indicating head vs discharge is shown to ascertain suitable minimum head of water required to be maintained in the trough to keep the wood wool pads always wet.

Experiments were conducted for three sizes of holes in the trough namely 2mmϕ, 2.5mmϕ and 3mmϕ. The first size hole displayed a high head in the trough. At the same time the holes were getting blocked with sediments in the water. On the other hand the head in the trough becomes very low for the 3.175mmϕ holes. Thus, there is no uniform water discharge over the lenth of the trough. Hence, finally 33 numbers of 2.5mmϕ hole size per meter was selected and recomended for instillation in the evaporative cooling system. The results are represented graphically in figure 5.4., and are also shown in table 5.4.

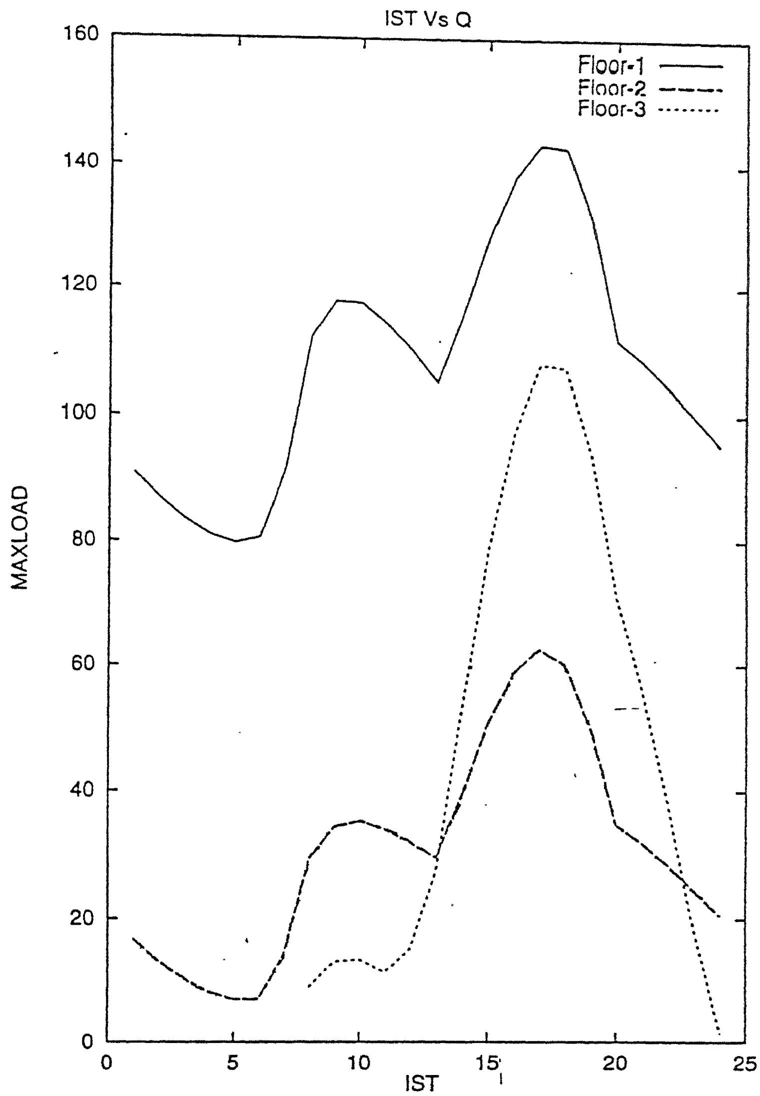

Figure 5.3: Variation of cooling load for May

5.3 Tabulated Results

The cooling loads for different floors of the triple storey building are given in Table 5.2 representing the maximum values. The electric load for each floor was taken as 40% of the connected load. The cooling load is found to be maximum in the month of May for which the air flow rates of the evaporatively cooled air required per floor of the building under

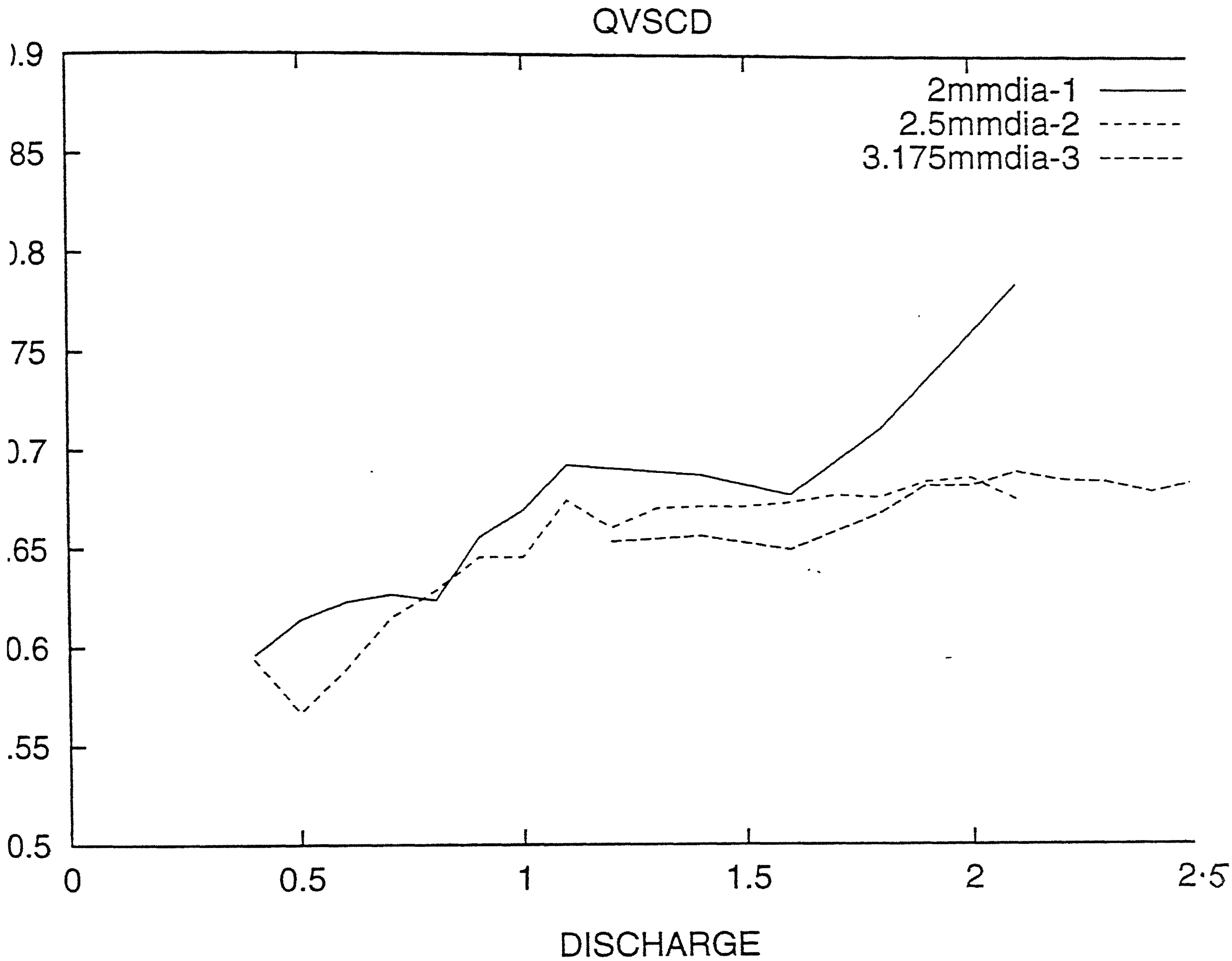

Figure 5.4: Cd Vs Discharge

study were evaluated.

Table 5.1: Maximum Cooling Loads for Ground Floor and Total Load

Month	Structload	Ventload	Glassload	Elecload	Occuload	Total
April	26.32	3.06	0.29	94.96	6.9	131.53
May	38.08	3.03	0.59	94.96	6.9	143.56
June	36.68	0.00	0.66	94.96	6.9	139.20
July	20.65	1.74	0.08	94.96	6.9	124.33
Aug	25.37	7.17	0.29	94.96	6.9	134.69
Sept	27.41	6.20	0.29	94.96	6.9	135.77
Oct	12.64	2.92	0.04	94.96	6.9	117.46

Table 5.2: Maximum Cooling loads for First Floor and Total Load

Month	Structload	Ventload	Glassload	Elecload	Occuload	Total
April	16.56	4.08	4.15	18.19	9.2	52.90
May	26.05	4.04	4.36	18.91	9.2	62.57
June	24.59	0.42	4.19	18.91	9.2	57.31
July	12.29	2.62	3.78	18.91	9.2	46.80
Aug	16.14	9.56	3.91	18.91	9.2	57.73
Sept	17.64	8.27	3.46	18.91	9.2	57.48
Oct	5.46	3.89	1.39	18.91	9.2	38.85

Table 5.3: Maximum Cooling loads for Second Floor and Total Load

Month	Structload	Ventload	Glassload	Elecload	Occuload	Total
April	86.91	1.79	4.15	1.46	4.02	98.33
May	96.17	1.77	4.36	1.46	4.02	107.78
June	95.85	0.00	2.91	1.46	4.02	104.24
July	82.27	1.02	2.20	1.46	4.02	90.97
Aug	83.22	4.18	3.91	1.46	4.02	96.79
Sept	82.90	3.62	3.46	1.46	4.02	95.46
Oct	65.22	1.70	1.39	1.46	4.02	73.79

5.3.1 Experimantal Results

The experimental results are tabulated for holes of different diameters. The measured flow rates for different heads of water in the trough are presented in Table 5.4 to 5.6. The discharge coefficient calculated from equations shows an increasing trend. The expression for Cd for different

diameters are given in equations. They can help calculate the discharge coefficient for the detirmination of the water required for circulation for the evaporative cooler pads. Hence, the total flow rate of water required for a given size of evaporative cooler pad can be estimated in this way.

The expression for the coefficient of discharge is given by:

- For 2mm ϕ hole size:

$$C_d = 4.3433 \times \frac{Q}{\sqrt{H}} \tag{5.1}$$

- For 2.5mm ϕ hole size:

$$C_d = 2.7802 \times \frac{Q}{\sqrt{H}} \tag{5.2}$$

- For 3.175mm ϕ hole size:

$$C_d = 1.7237 \times \frac{Q}{\sqrt{H}} \tag{5.3}$$

where,
Q = the discharge in U.S.gallons/min.
H = the head in the trough in mm.
C_d = the coefficient of discharge.

Table 5.4: Experimental results for 2mm ø hole Size

Sl No	Discharge(US Gal/min)	Head (mm)	Coeff.of Disch.(Cd)
1.	0.4	8.5	0.614
2.	0.5	12.5	0.614
3.	0.6	18	0.614
4.	0.7	24	0.620
5.	0.8	30	0.634
6.	0.9	36	0.651
7.	1.0	41.5	0.674
8.	1.1	52	0.662
9.	1.2	57	0.690
10.	1.4	78.5	0.686
11.	1.6	105	0.678
12.	1.8	121	0.717
13.	2.0	130	0.762
14.	2.1	135	0.78

Table 5.5: Experimental results for 2.5mm ø hole Size

Sl No	Discharge(US Gal/min)	Head (mm)	Coeff.of Disch.(Cd)
1.	0.4	3.5	0.594
2.	0.5	6.0	0.567
3.	0.6	8.0	0.589
4.	0.7	10	0.615
5.	0.8	12.5	0.629
6.	0.9	15	0.646
7.	1.0	18.5	0.646
8.	1.1	20.5	0.675
9.	1.2	25.5	0.660
10.	1.4	33.5	0.672
11.	1.6	43.5	0.674
12.	1.8	54.5	0.677
13.	2.0	65.5	0.687
14.	2.1	74.5	0.676

5.4 Density of Wood Wool Pads

The ASHRAE Handbook of Equipment [7] gives that the density of wood fibers used for an evaporative cooling system should vary between

Table 5.6: Experimental results for 3.175mm ø hole Size

Sl No	Discharge(US Gal/min)	Head (mm)	Coeff.of Disch.(Cd)
1.	1.2	10	0.654
2.	1.4	13.5	0.657
3.	1.6	18	0.650
4.	1.8	21.5	0.669
5.	1.9	23	0.682
6.	2.0	25.5	0.682
7.	2.1	27.5	0.690
8.	2.2	30.5	0.686
9.	2.3	33.5	0.684
10.	2.4	37	0.680
11.	2.5	39.5	0.685

1.5kg/m^2 to 2.0kg/m^2 for a thickness of 5cm. of the pad surface area. The pad thickness should vary between 30mm and 55mm. In the present case 50mm pad thickness has been recomended due to the large size of pad. An experimental result of reference [21] shows an optimum density of 1.97 kg/m^2 and 2.5kg/m^2 of the pad surface area. These values are even higher than those recomended by ASHRAE. Hence, it would render an increased pressure drop and hence reduce the air handling capacity of the fan. In the present case, however, the wood wool density of 1.5kg/m^2 of the pad surface area based on the practical considerations and experiment is found to be appropriate for the evaporative cooling system recomended for the building. The photographs of the same are attached below.

5.5 Axial-Flow Fan

On reffering to the catelogues of standard manufactures the following axial-flow fans were recomended for instillation in the evaporative cooling systems of each floor. The head for fans in millimeters of water are recomended on the basis of the total pressure drop required for the air supply, evaporative cooling devices, ducts, etc. The details of the same

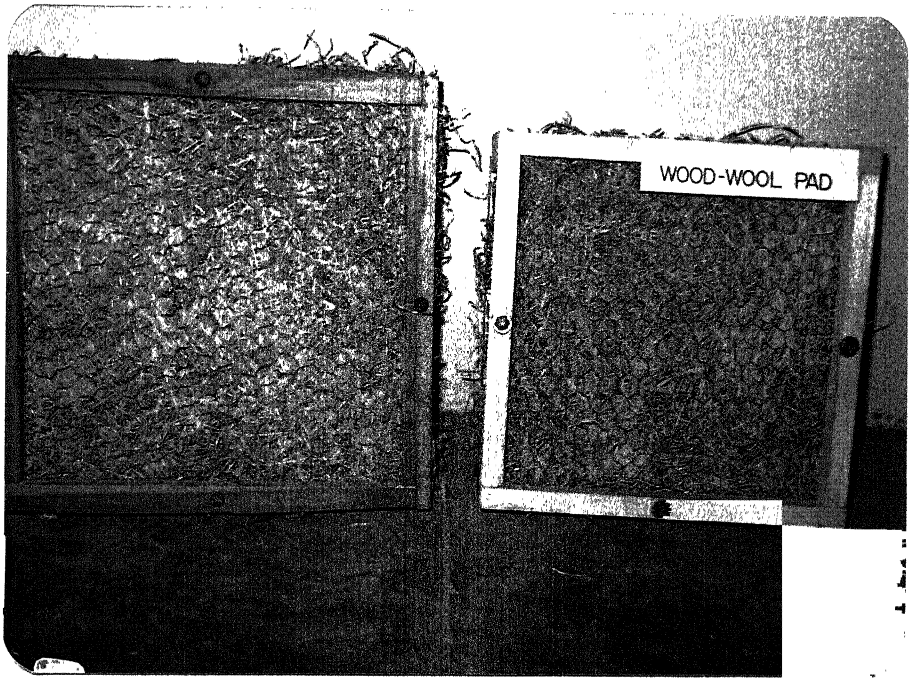

Figure 5.5: Photographs of recomended Wood Wool Pads

are given below:

- Ground Floor:
  (a) Discharge: 80,000 m^3/h
  (b) Head: 30 mm of water.
- First Floor:
  (a) Discharge: 40,000 m^3/h
  (b) Head: 25 mm of water.
- Second Floor:
  (a) Discharge: 14,400 m^3/h
  (b) Head: 20 mm of water.

5.6 Duct Design and Layout

As explained in chapter4 the duct layout and size were calculated with the help of a computer program and the equal friction charts for air and GI sheets. The layout and sizes are as indicated in the drawings attached.

5.7 Water Distribution

Knowing all the parameters of the trough from the experiments conducted and refering to the catelogues of standard manufactures the following was recomended

- Ground Floor:
  (a) Rating: 0.75 kW
  (b) Discharge: 5000 L/h
  (c) Head : 6m of water
- First Floor:
  (a) Rating: 0.37 kW

(b) Discharge: 2400 L/h

(c) Head: 6m of water

- Second Floor:

  (a) Rating: 0.18 kW

  (b) Discharge: 1200 L/h

  (c) Head: 6m of water

5.8 Design of Wood Wool Pads

- Ground Floor: The pads have two straight sections with curved pads at the back as given in Fig.5.6.
- First Floor: Two straight pads each 2m long having a curved length of 3.3m length, each 2m high and 50mm thick having the mass of woodwool as 1.5kg/m^2 of the pad surface area. The details of the same are shown in Fig 5.7.
- Second Floor: The pads have two straight pads 0.5m long having a semicircular pad with radius 0.75m, each being 1.5m high as shown in Fig 5.8.

The evaporative cooling systems for all the three floors have been placed on the roof of the triple storey manufacturing shop due to the space restrictions imposed within the factory. A detailed plan view and elevation of the systems are given in the drawings attached. It is worth mentioning that the factory is located in an area where ammonia vapours from a fertilizer factory in the vicinity of the building under study causes significant nuisance. The evaporatively cooled air supply does away with this problem and provides a better quality of air for the users inside the workshop.

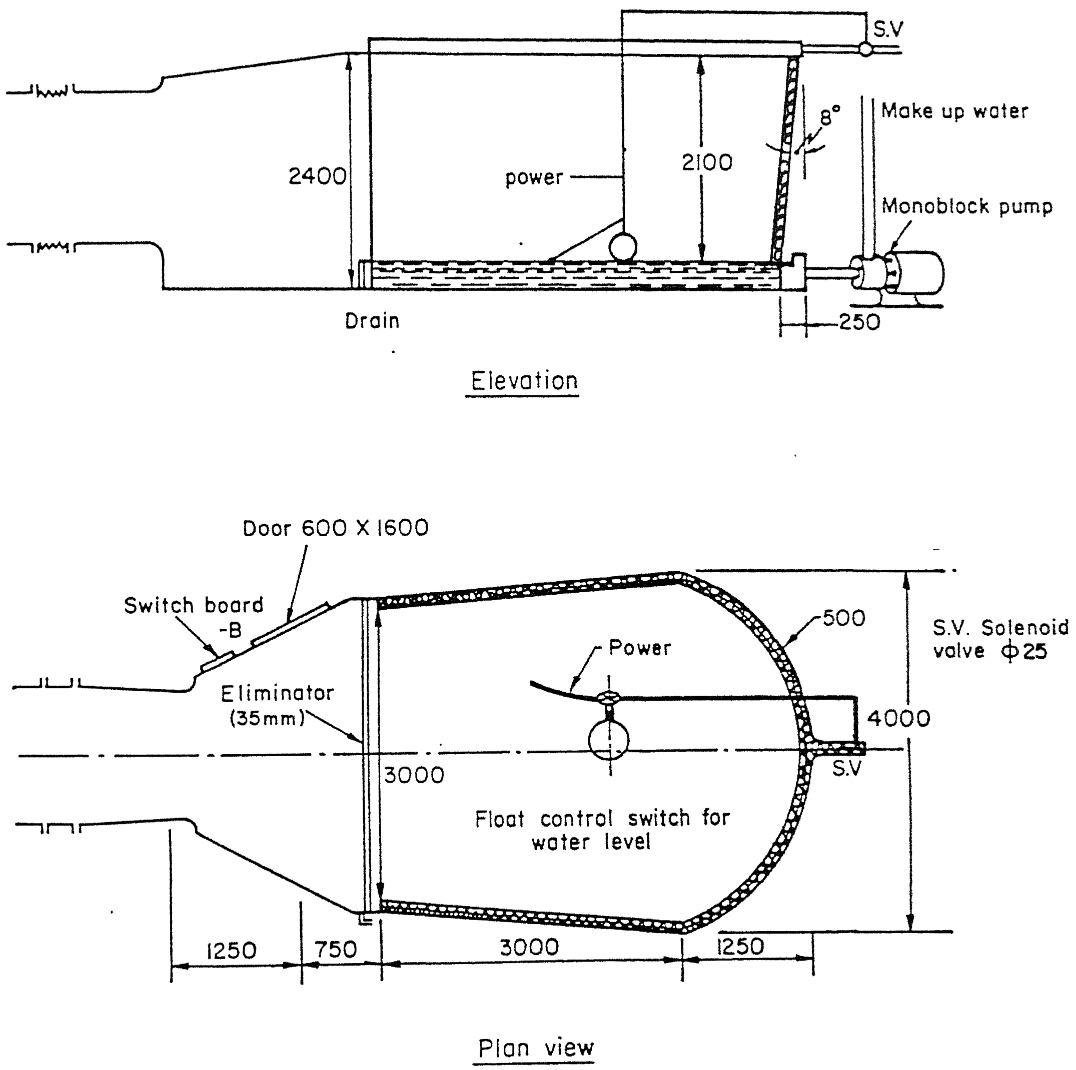

Figure 5.6: Evaporative cooling system for ground floor

5.9 Design of Water Distribution System

A layout of the water distribution system proposed are given in the layout of the plan view depicting the cooling systems placed on the roof

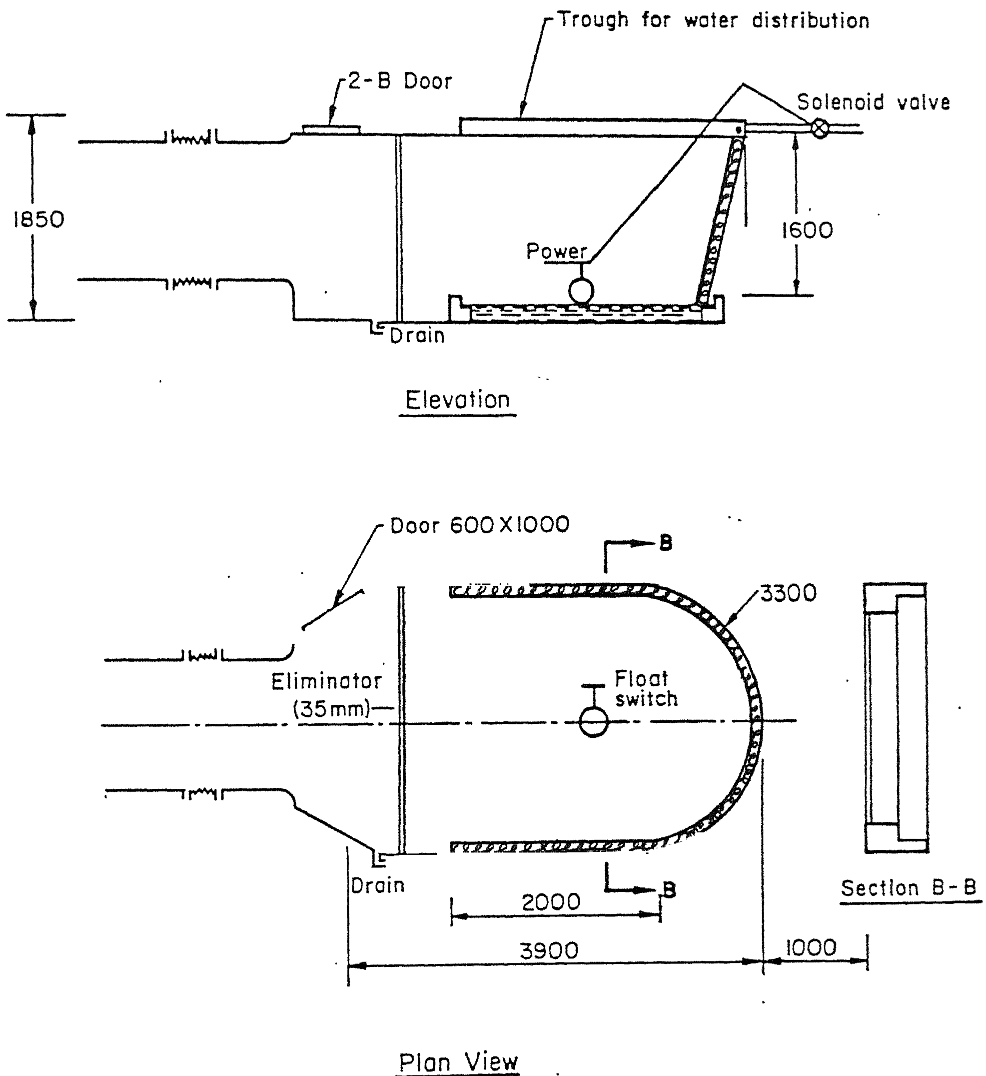

Figure 5.7: Evaporative cooling system for first floor

of the manufacturing shop. The water supply is divided along 3 paths for each evaporative cooler. Two water supply pipes for the straight pads and one for the curved pads. This arrangement helps operate the

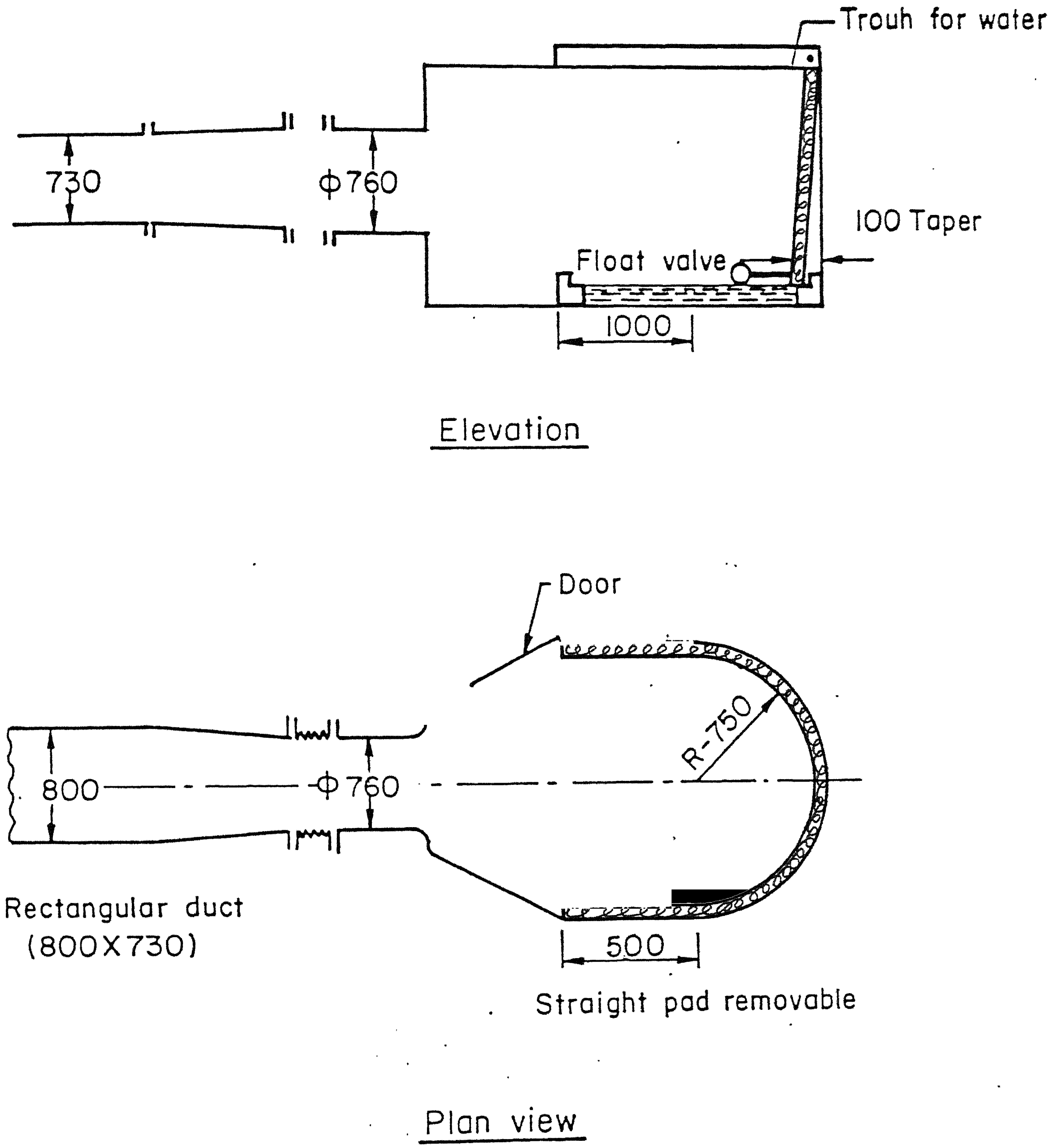

Figure 5.8: Evaporative cooling system for second floor

Figure 5.9: Plan view and elevation of the evaporative cooling systems

evaporative system as usual with the water distribution to all pads in the summer season. But, in the rainy season when the humidity of ambient air is very high Kanpur (about 98%), the water distribution to the pads can be restricted by stopping the water supply to any one of the pads.

To get rid of continuous operation of the pump, an electronic timer is introduced. This arrangement renders the following advantages :

- Wear and tear of pump reduces drastically.
- Electric energy consumption reduces to about $\frac{1}{5}^{th}$ of the normal electric energy consumption.
- Sound of the operating pump is reduced.
- The motor runs always cool and hence the burning of the coil winding in the motor is dispensed with.

Chapter 6

Conclusions and Suggestions

6.1 Conclusions

The following are the salient features of the present work :

(a) A computer programme has been developed to predict the total cooling load on the basis of the hourly temperature variation based on T_{max} and T_{min} for a given day. The versality from the present approach is supported from the fact that only T_{max} and T_{min} values are needed in the computation and the same are available in several data books [5,16,17] for many places. But the hourly variation is available only at selected meteorological centers.

(b) The cooling load for the summer months is found to be maximum in the month of May. This is in tune with the recommended month for practice engineers by the standard texts.[10,14,15]

(c) The ventilation air is eatimated on the basis of energy balance between the heat gained by the building and the cooling to be produced by the evaporatively cooled air.

(d) The evaporative cooling systes are provided with eliminators between the pad and the axial-flow fans to avoid water droplets from entering the ducts and the conditioned space through the

ducts.

(e) All bends, in the ducts have been provided with vanes to reduce pressure losses and dampers to control the air supply at desired rates. To reduce pressure losses a gradual expansion and contraction is recomended for the duct whenever there is a change in section

(f) The layout of the duct work was so designed that all occupants get the required volume of fresh evaporatively cooled air at the same time a simple network of ducts are proposed to help in its construction and instillation.

(g) Solenoid valves have been provided in the water distribution system to help cnserve the wastage of water. The use of electronic timers for the pumps used for make up water (manufactured by Mr Alok Prasad, at I.I.T.Kanpur) have been proposed which also help in conserving energy.

(h) The straight pads and the curved pads are connected separately to water supply pipes so that in the summer season the evaporative system will operate with the water distribution to all pads but in the rainy season when the humidity of ambient air is high, the water can be supplied to only one or two pads only. In this way the humidity can be controlled to some extent.

(i) To avoid the instillation of an oversized pump for ground floor systems a stand by pump has also been recomended, which will be supplying water to the curved pads only.

(j) The manufacturing shop is located in the industrial area in Kanpur and it was observed that there was a predominant odor of ammonia gas in the environment. The ammonia will be dessolved in the water flowing down the evaporative cooler pads and hence, the system will supply better quality air to the work place.

6.2 Scope for Future Work

(a) This computer programme to calculate the cooling load is applicable to the buildings located in the Northern Hemisphere. This program can be suitably modified for the buildings located in the Southern Hemisphere by changing the subroutine which calculate the solar radiation intensity.

(b) The problem can be extended to one used for all the year round air conditioning by the use of hybrid airconditioning system.

(c) Economic analysis can be done for the evaporative cooling system to find initial investment and running cost, though it can be concluded that it will less than the cost of installing and running a central airconditioning system for the same building.

(d) As discussed in chapter 1 more publicity should be given to the evaporatively cooled system because

   i. There is substantial energy and cost savings.

   ii. No chloroflurocarbon usage.

   iii. Reduced CO_2, i.e. no ozone depletion, and power plant emissions,

   iv. Improved indoor air quality,

   v. It is easily intergrated in built up systems,

   vi. Potential for significant local fabrication and employment.

(e) A more generalised programme needs to be developed which may design all ducts after taking input data.

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Appendix A

Percentage of the Daily Range

Table A.1: Percentage of the Daily Range

time,h	%	time,h	%	time,h	%	time,h	%
1	87	7	93	13	11	19	34
2	92	8	84	14	3	20	47
3	96	9	71	15	0	21	58
4	99	10	56	16	3	22	68
5	100	11	39	17	10	23	76
6	98	12	23	18	21	24	82

Appendix B

Temperature and Relative Humidity of Outside Air

Table B.1: Hourly Outside Temperature (^oC) for Kanpur

time,h	Apr	May	June	July	Aug	Sep	Oct
1	28.2	31.5	33.2	30.9	31.6	31.3	22.0
2	27.0	30.7	33.0	30.4	31.5	31.0	21.6
3	26.7	30.0	32.3	29.9	31.4	30.4	21.5
4	25.2	29.4	31.7	29.8	30.8	30.7	18.8
5	25.2	29.8	31.4	29.4	30.8	30.9	19.8
6	25.3	29.8	31.6	29.4	30.9	30.4	20.2
7	25.2	30.4	31.8	29.5	31.4	30.5	20.3
8	27.4	32.2	32.5	30.2	31.7	32.4	22.5
9	30.6	36.2	35.3	31.8	33.9	33.7	25.9
10	31.6	37.5	36.4	33.0	34.3	34.3	26.9
11	33.1	39.5	37.9	32.8	34.8	35.1	28.3
12	34.6	40.4	38.8	33.7	31.9	36.0	29.6
13	35.9	41.5	40.2	34.3	35.9	36.1	29.9
14	36.1	42.0	40.8	34.2	36.0	36.4	32.3
15	36.3	41.8	41.1	34.8	36.0	36.6	32.6
16	36.6	42.4	41.1	34.9	36.6	35.9	32.9
17	36.1	41.8	41.3	35.2	34.8	35.3	30.2
18	35.2	40.2	40.3	34.5	34.4	34.3	27.7
19	33.4	36.3	38.2	33.5	33.6	33.9	26.1
20	32.3	36.9	36.8	32.8	33.1	33.2	25.3
21	31.2	34.8	35.5	32.1	32.9	32.7	24.1
22	30.5	33.5	35.0	31.3	32.6	32.4	23.9
23	29.6	32.7	34.0	31.3	32.1	32.2	22.9
24	28.9	32.2	33.7	31.1	31.7	31.9	22.6

Table B.2: Hourly Outside Relative Humidity (%) for Kanpur

time,h	Apr	May	June	July	Aug	Sep	Oct
1	53.3	44.7	35.3	56.3	88.7	82.3	78.0
2	55.3	44.7	35.7	60.7	89.3	86.3	83.3
3	58.3	45.0	35.7	62.0	91.7	85.7	84.3
4	60.7	46.7	35.0	64.0	92.0	85.7	84.0
5	63.3	48.0	34.7	65.3	90.3	84.0	87.7
6	66.0	48.0	36.7	62.0	88.7	80.0	88.0
7	68.0	45.7	34.0	63.3	74.7	76.3	84.7
8	62.3	39.3	31.7	68.0	81.0	70.0	71.3
9	54.0	37.3	33.0	56.7	78.7	60.0	56.7
10	44.0	33.0	29.7	61.7	77.0	54.7	50.0
11	40.0	29.3	26.0	54.3	69.7	54.0	46.3
12	38.0	29.3	24.3	45.0	66.0	51.7	40.3
13	35.0	28.3	22.3	43.7	63.3	51.0	37.0
14	33.0	27.3	20.0	38.7	60.0	50.7	34.7
15	32.3	26.0	18.0	36.0	60.7	49.0	30.7
16	30.3	26.0	15.7	36.3	61.7	53.3	35.7
17	29.3	17.0	16.7	37.3	62.7	58.7	41.0
18	31.3	28.3	19.3	41.7	68.7	69.3	50.0
19	36.7	30.7	23.0	44.3	71.0	72.0	53.3
20	41.0	35.0	25.7	44.0	77.7	75.0	65.3
21	43.3	37.7	27.7	50.7	84.0	74.0	71.3
22	47.7	41.7	30.7	52.3	85.7	80.0	72.0
23	49.3	44.0	32.7	54.0	85.3	81.7	73.3
24	50.0	44.0	33.0	54.3	85.3	80.3	75.3

Appendix C

Constants for Solar Radiation Calculation

Table C.1: Constants for Solar Radiation Calculation

Month	Equation of Time min.	A W/m^2	B (Dimensionless Ratios)	C
Jan	-11.2	1230	0.142	0.058
Feb	-13.9	1214	0.144	0.060
Mar	-7.5	1185	0.156	0.071
Apr	1.1	1135	0.180	0.097
May	3.3	1103	0.196	0.121
Jun	-1.4	1088	0.205	0.134
July	-6.2	1085	0.207	0.136
Aug	-2.4	1107	0.201	0.122
Sep	7.5	1151	0.177	0.092
Oct	15.4	1192	0.160	0.073
Nov	13.8	1220	0.149	0.063
Dec	1.6	1233	0.142	0.057

A = Apparent Solar irradiation at zero air mass , (W/m^2)

B = Atmospheric extinction coefficient

C = Diffuse radiation factor

Appendix D

Properties of Ordinary Glass

Table D.1: Properties of Ordinary Glass for Direct Radiation

Angle of Incidence	0^o	10^o	20^o	30^o	40^o	50^o	60^o	70^o	80^o	90^o
Transmissivity, τ_D	0.87	0.87	0.87	0.865	0.86	0.84	0.79	0.67	0.42	0.00
Absoptivity, α_D	0.05	0.05	0.05	0.055	0.06	0.06	0.06	0.06	0.06	0.00

For diffuse radiation, regardless of the angle of incidence

$\tau_d = 0.79$

$\alpha_d = 0.06$

Appendix E

Input Data

Table E.1: Input Data For Ground, First and Second Floors.

Sl.No.	Building Parameter	Ground Floor	First Floor	Second Floor
1.	lat, long (°)	27, 82	-do-	-do-
2.	Absorptivity of structure	0.8	-do-	-do-
3.	Outside Velocity (m/s)	3.194	3.194	3.194
4.	Inside Velocity (m/s)	1.2	1.2	1.2
5.	No. of Walls	4	4	4
6.	Volume of Space (m^3)	2638	1929	1929
7.	No.of Elec.Appl	10	2	1
8.	Their Rating (kW)	20	10	1
9.	No. of Lights	85	62	30
10.	Their Rating (kW)	0.2	0.2	0.4
11.	No. of Occupants	60	80	35
12.	East Wall Thick (m)	0.139	0.139	0.139
13.	North Wall Thick (m)	0.139	0.139	0.139
14.	West Wall Thick (m)	0.139	0.139	0.139
15.	South Wall Thick (m)	0.139	0.139	0.139
16.	East Wall Area(m^2)	202	148	148
17.	North Wall Area(m^2)	64	47	47
18.	West Wall Area(m^2)	202	148	148
19.	South Wall Area(m^2)	64	47	47
20.	Windows East Wall	2	2	4
21.	Windows North Wall	1	1	1
22.	Windows West Wall	2	2	4
23.	Windows South Wall	1	1	1
24.	Data for Roof	in shade	in shade	not shaded
25.	Roof Thickness (m)	0.174	0.174	0.174
26.	Area of Roof(m^2)	533	533	533